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A Final Report

"Evaluation of Truck Tire Rolling Resistance: Bearing Losses and Road Surface Effects on Rolling Resistance - Phase II"

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D.S.S. File No. 17ST.T8201-8-8501
D.E. File No. C-360

January 1982

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ABSTRACT

A project to design, fabricate and test a rig for measuring the over-the-road truck tire rolling resistance and bearing moment losses was undertaken by Davis Engineering Limited for Transport Canada. Phase I of the project, which included the design and fabrication of the force bridge and initial on-road validation tests, was conducted between July, 1978 and July, 1979. The present Phase II was initiated in February, 1980, to calibrate the force bridge and TARR system, automate the data acquisition system, design and install additional control and parameter monitoring systems, and to acquire and test chosen truck tires. The results of a short over-the-road test program indicated that further development of the rig may be required in some areas, but the overall performance of the rig was encouraging and lead to optimism towards a successful completion of the project objectives. In conjunction with the comments and recommendations of the project Steering Committee, recommendations have been made to refine certain aspects of the rig followed by further verification testing leading to a full calibration and road test program.

1.0 INTRODUCTION

1.1 Background

The energy losses associated with the rolling resistance of large truck tires and their wheel bearings account for a significant portion of the total power requirements of a truck trailer combination. These losses can be quantified for some ideal conditions in laboratories and through theoretical considerations. However, the effects that a wide range of tire designs, road surface conditions, ambient temperature conditions, and bearing and grease designs have on the losses are difficult if not impossible to determine through laboratory testing or by theoretical consideration without verification by actual over-the-road test results. For these reasons, a need was identified to develop a device that could measure over-the-road losses for a wide range of parameters.

With the above in mind, Davis Engineering Limited went into contract with the Strategic Studies Branch of Transport Canada to design, develop and test a rig for measuring the over-the-road tire rolling resistance and wheel bearing moments of large trucks.

A steering committee was formed to provide input and assistance to the project. This committee comprised representatives from the following government and industry sectors: Strategic Studies Branch (TC), Road Safety Branch (TC), Transportation Industries Branch (ITC), Conservation & Renewable Energy Branch (EMR), U.S. DOT Heavy Truck & Bus Program, Ontario Ministry of Transportation & Communications, Canadian Trucking Association, Michelin Tires (Canada) Ltd., Canadian Rubber Association (Tire Committee), Department of Supply and Services Canada, Canada Department of Industry, Trade and Commerce, Calspan Corporation, DEW Engineering and Firestone.

Phase I of the project was conducted between July 1978 and July 1979. That phase consisted of the basic design, fabrication and validation testing of the rig. The results of those preliminary tests indicated that the rig had good sensitivity and warranted further development.

Phase II of the project, the current phase, commenced in February, 1980, and is documented in this report.

1.2 Objectives

Following the successful demonstration of the test rig developed in Phase I, Phase II was initiated as a continuation of the project. Alterations were made to the scope of the project while in progress, as discussed in Section 3.1. Additional scope resulted from recommendations made by the Steering Committee. A reduction in scope resulted from problems encountered in course of the project.

The resulting objectives carried out are as follows:

(a) Calibration - design and fabricate calibration rig
- calibrate force bridge and reduce data

(b) Data Acquisition System
- design, develop and install on-board data acquisition system including in-cab display

(c) Design, Fabricate and Install Additional Equipment
- tire inflation pressure control system
- tire contained air temperature measurement system
- define constant temperature enclosure requirements

  - road surface measurement system
  - fifth wheel for speed measurement
  - air temperature measurement system

(d) Error Analysis
  - develop analytical procedures for performing system error analysis

(e) Baseline Test Tires at Laboratory (Calspan)

(f) Overload and Pressure Loss Safety System

(g) Assemble System

(h) Perform Validation Tests with Complete System.

2.0 DESCRIPTION OF TEST RIG

The principle of operation and detailed description of the test rig was provided in the final report for Phase I(1)*. Hence, only a brief description of the rig is provided in the following.

The measurement of rolling resistance and bearing moment is achieved by an instrumented force bridge mounted between the test axle containing two sets of dual wheels and a towed 13.7 m (45 ft.) semi-trailer as shown in Figure 1. A second quick lift tag axle is used for raising the test axle off the road surface during transportation and also for braking. The axle loading is varied by changing the trailer ballast. A shield is mounted to reduce aerodynamic effects on the test tires.

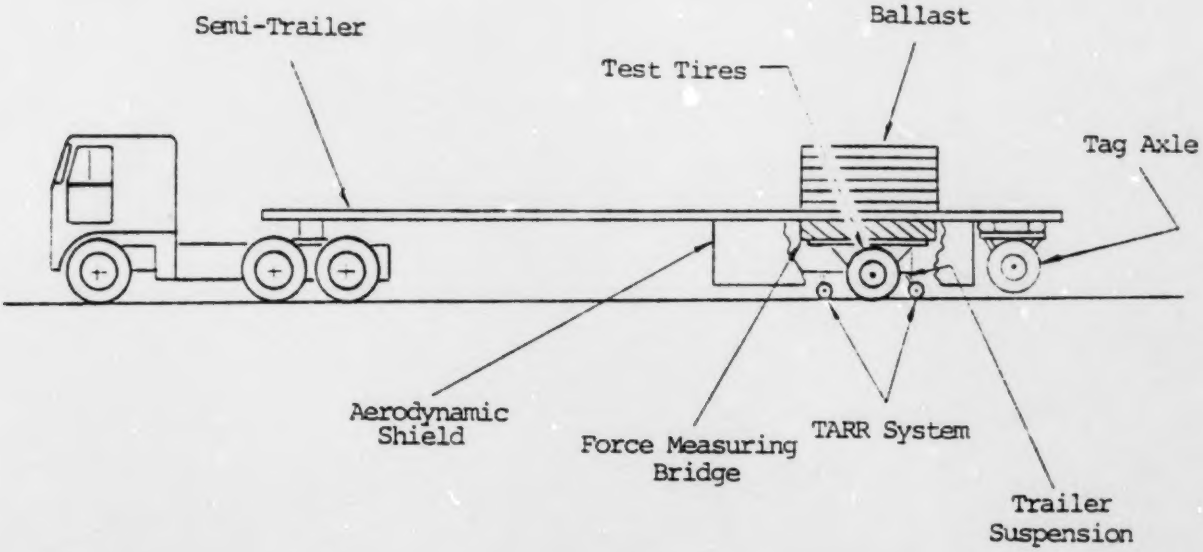
Semi-Trailer
Ballast
Test Tires
Tag Axle
Aerodynamic Shield
Force Measuring Bridge
TARR System
Trailer Suspension

Figure 1 - Force Measuring Trailer

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* Number in bracket refers to the reference given in Section 5.0

In order to limit the longitudinal travel of the axle, a solid rubber suspension manufactured by Hendrickson was chosen. A Trailmobile axle (capacity = 15,500 kg.) compatible with the suspension was chosen as the test axle.

The force bridge is the key to the entire apparatus. The basic arrangement of the force measuring bridge is illustrated in Figure 2. The device consists of two interconnected platforms, one rigidly attached to the trailer bed. The two platforms are connected by a series of load cells and flexures. The flexures are rigid along one axis and flexible on the other two orthogonal axes. By judiciously selecting the location and orientation of the load cells and flexures, one can measure all of the forces necessary to determine rolling resistance, bearing torque, and vertical and lateral loads, while adequately fastening the suspension to the semi-trailer.

For rigidity, the force bridge is constructed of one inch thick steel plates. To avoid potential slippage, the joints are completely welded.

The nominal ratings of the load cell/flexure assemblies are as follows:

Vertical:	9,000 kg (each)
Lateral:	2,300 kg
Lower Longitudinal:	1,400 kg (each)
Upper Longitudinal:	2,300 kg

To minimize load cell output errors due to temperature fluctuations, it is intended that the force bridge be surrounded by a constant temperature enclosure, controlled to the nominal calibration temperature (20°C).

FORCE BRIDGE

LOAD CELLS

C, D, E — Longitudinal Force and Bending Moment
A, B — Vertical Force
F — Lateral Force

ANGLE SENSORS

X — Roll
Y — Pitch
Z — Road / Tire Angle

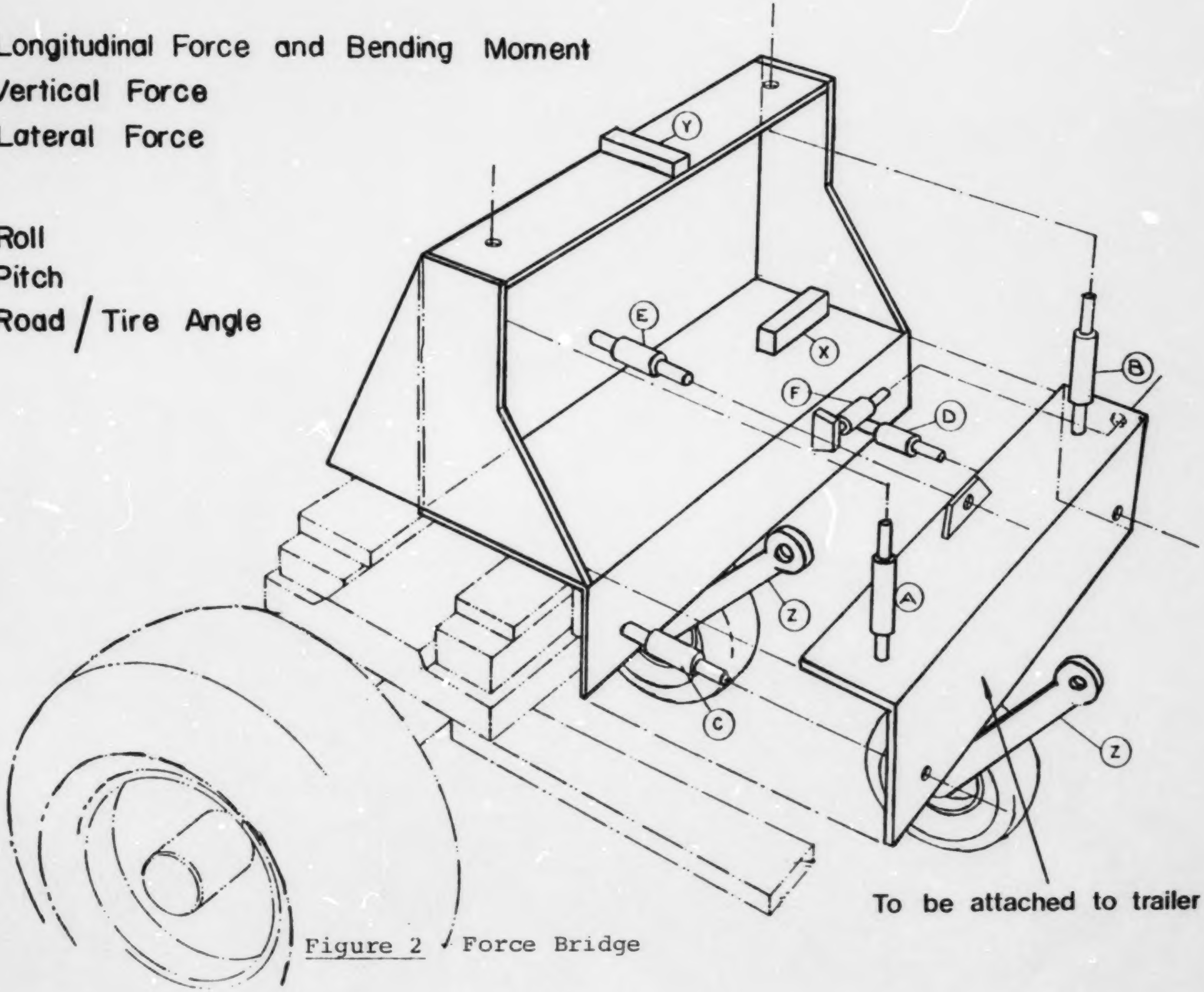

Y
E
X
F
D
B
A
Z
C
Z
To be attached to trailer

Figure 2 / Force Bridge

The force bridge pitch and roll angles are measured with respect to the horizontal by two inclinometers ($\pm 3^{\circ}$) mounted on machined surfaces of the force bridge.

A tangent angle rolling radius (TARR) system (illustrated in Figure 3) is used to measure the angle of the force bridge with respect to the road surface as well as the rolling radius of the test tires.

The TARR system uses a pair of road following wheels which are in contact with the road surface 55.9 cm in front and 55.9 cm behind the measuring axle. The motions of the wheels relative to the force bridge are detected through LVDT's (Linear Variable Differential Transformer) having a ±5.1 cm travel and from these measurements, both angle and average rolling radius of the axle wheels are determined.

The wheels used were developed for road profilometers and are specially tuned to damp out vibration when held against the road with a force of approximately 670 N. The wheels are lifted clear of the road when not traversing a test section and this is accomplished using solenoid controlled dual acting air cylinders.

In order to allow for large variations in tire radius and still have a ±5.1 cm measuring range on each LVDT, the TARR system was designed such that the complete unit could be blocked with 2.5 cm shims.

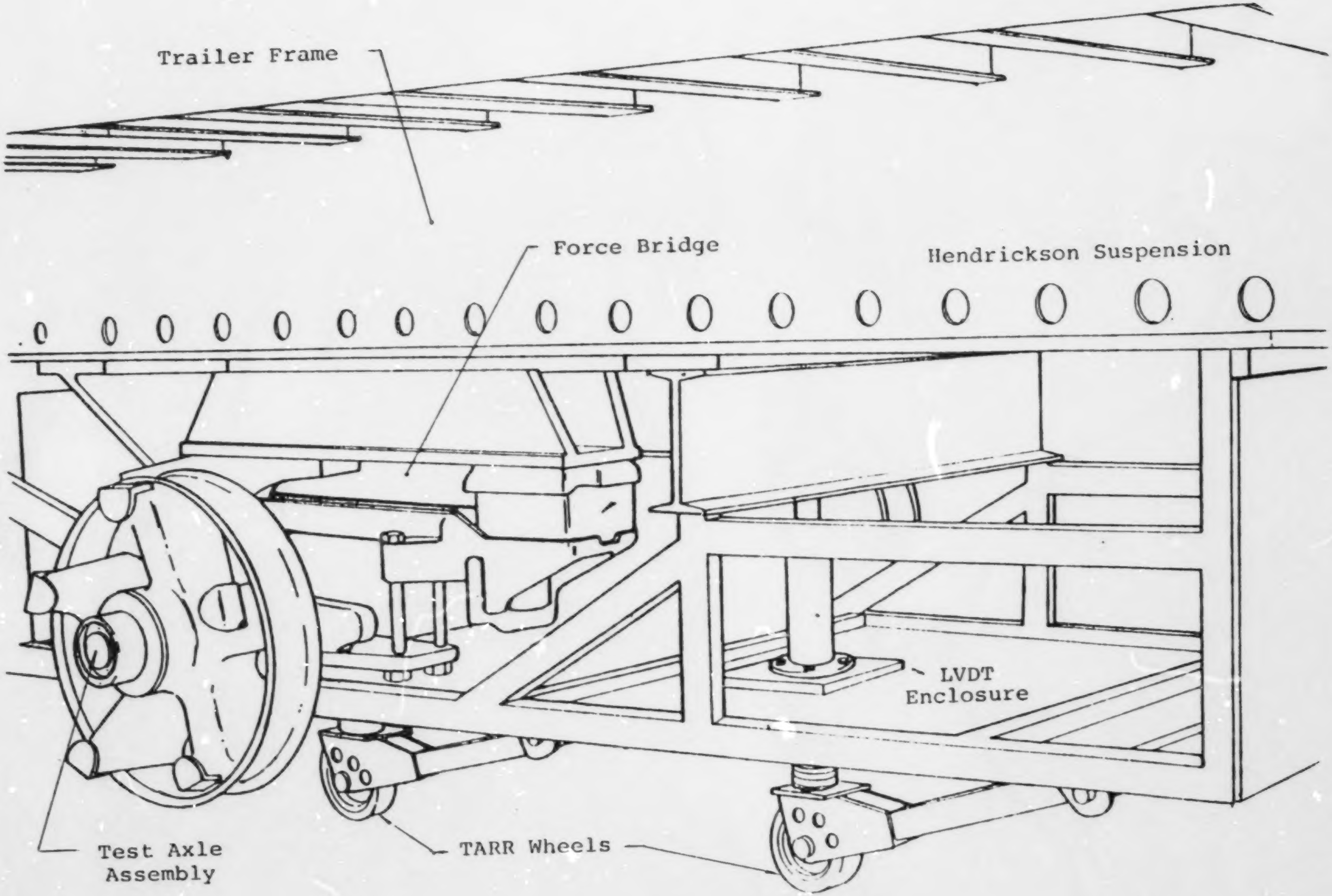

Trailer Frame
Force Bridge
Hendrickson Suspension
LVDT Enclosure
Test Axle Assembly
TARR Wheels

Figure 3 - System Assembly

3.0 SUMMARY OF PROJECT ACTIVITIES

3.1 Change in Scope

Unforeseen difficulties encountered primarily during the calibration of the force bridge, additional scope relating to safety aspects of the rig, and general inflation effects lead to a change in scope of the project in order to limit potential budgetary and schedule overruns. Some activities were consequently deferred for future consideration.

3.1.1 Difficulties Encountered During Force Bridge Calibration

(a) Load Spring Failure

A load spring failed prematurely during the calibration trials resulting in the failure of four force-bridge flexures and a complete destruction of all prior calibration effort to that date. Although a new spring was provided under warranty from the supplier with a reasonable delivery time, the replacement flexures had a seven month delivery time and were costly. In addition, considerable effort was consumed in dismantling the force bridge and verifying that the other components, including the load cells, had not been damaged. Other methods of loading the force bridge were explored to prevent a possible recurrence of a spring failure. Eventually the vertically loading springs were eliminated from the loading procedure.

(b) Load Alignment and Cross-Coupling

The initial calibration trials showed considerable cross-coupling to orthogonally applied loads. For example, the vertical load cells responded linearly to applied vertical loads, but the orthogonally located load cell responses were erratic. In none of these trials was there any indication that the force bridge itself was responding erratically, but rather this was due to applied load misalignment or to undesirable loads being applied together with the desirable loads. After considerable experimentation, the following aspects were noted and were corrected to reduce the excursions of the cross-couplings from a linear response.

i. Rocking action occurred between the replacement solid suspension, force bridge, and the axle.

ii. The axle was deforming vertically appreciably so that its end rotations in the vertical plane were significant and had to be taken into consideration.

iii. The jactuator (screw jack) shafts moved horizontally with the application of vertical loads causing misalignment.

iv. Due to frictive seizure of the bearing joints, the loading bars transmitted moments in addition to linear forces.

In order to correct the undesirable effects imparted by the above found distortions, the following precautionary measures were made to rectify the system.

- the solid suspension was shimmed and spot welded to prevent rocking and slippage (during the linearity check);

- the angular rotations of the axle were monitored and corrective measures taken during the load application; in this manner, the moments transmitted were minimized;

- a special guided sleeve was designed and fabricated so that the jactuator shaft was restrained from moving laterally and thus preventing misalignment; and

- where appropriate, bearing rod ends were replaced with pivotal joints.

3.1.2 Safety Features

To safeguard the force bridge from damage when in service, additional activities were carried out beyond the original scope, being:

- the warning and overload protection of the force bridge cells and flexures; and,
- a warning and protection for pressure loss in the tag axle actuation and in the inflation pressure control system.

These items were recommended by the Steering Committee for safety reasons.

3.1.3 Revised Scope Leading to Demonstration

The revised scope of the project comprised a demonstration of the force bridge fully calibrated to a chosen vertical payload and to utilize the automatic data acquisition system. Although the testing program was minimized, it was planned to obtain sufficient data for comparison of rolling resistance with laboratory results and for a preliminary assessment of bearing moment resistance and tire/road surface effects. From this experience, comments relating to future work were made and recommendations given for future improvements in the precision of recorded or derived results.

Those activities which were deferred for consideration after a demonstration of the calibrated force bridge included the following:

(a) static calibration over the full operation range of the force bridge and its incorporation in software programs;

(b) complete the constant temperature enclosure - for operating over a wide range of temperatures;

(c) complete the pulse generator (for measuring slip and phases between the two sets of dual wheels) installation;

(d) error analysis - utilize the developed theory to include the results of the calibration over the full dynamic range, which will lead to improvement of the definition of the results, performance, and establish confidence limits;

(e) road test program - a full program for three or more sets of tires and various bearings should be carried out after a calibration over the full dynamic range of the force bridge;

(f) assessment of the energy penalties with different tire/road surface interactions; and

(g) re-baseline tires - to be carried out after the full road test program to re-establish the rolling resistance values as a result of appreciable tire wear.

3.2 Progress Made on Activities

3.2.1 Calibration Trials and Data Reduction

3.2.1.1 Calibration Rig

The rig shown schematically in Figure 4 was designed and fabricated for calibrating the force bridge. It consists of a structural steel frame, suitable for mounting the force bridge and applying loads in the vertical, lateral and longitudinal directions as well as applying a mount about the axle. The legs contain adjustments for the purposes of levelling the force bridge and to apply roll and pitch rotations. The vertical, lateral, and longitudinal forces (for linearity check) are applied through a loading jactuator (screw jack) arrangement. Moments are applied by using a dead weight/pulley arrangement. The vertical and lateral loads are applied at the axle ends. The longitudinal loads are applied through the centroid of the three longitudinal load cells of the force bridge for the linearity check. In the actual calibration, the axle and longitudinal loads were applied with a dead weight pulley system. The applied loads in the vertical, lateral, and longitudinal directions were known from the calibrated load cells used in series with the applied forces except where dead weight was used, in which case no load cells were used in series. These load cells were calibrated on a dead weight facility at NRC. The applied moment was determined from the weight of the dead weights and the measured length of the moment application arm.

Angular alignments of the force bridge were measured with the inclinometers contained in the force bridge. Vertical alignment of the loads was facilitated by use of an inclinometer on the plate of the axle bearing sleeves. Other alignments were made by use of a grid on the floor and with surveyor's transits.

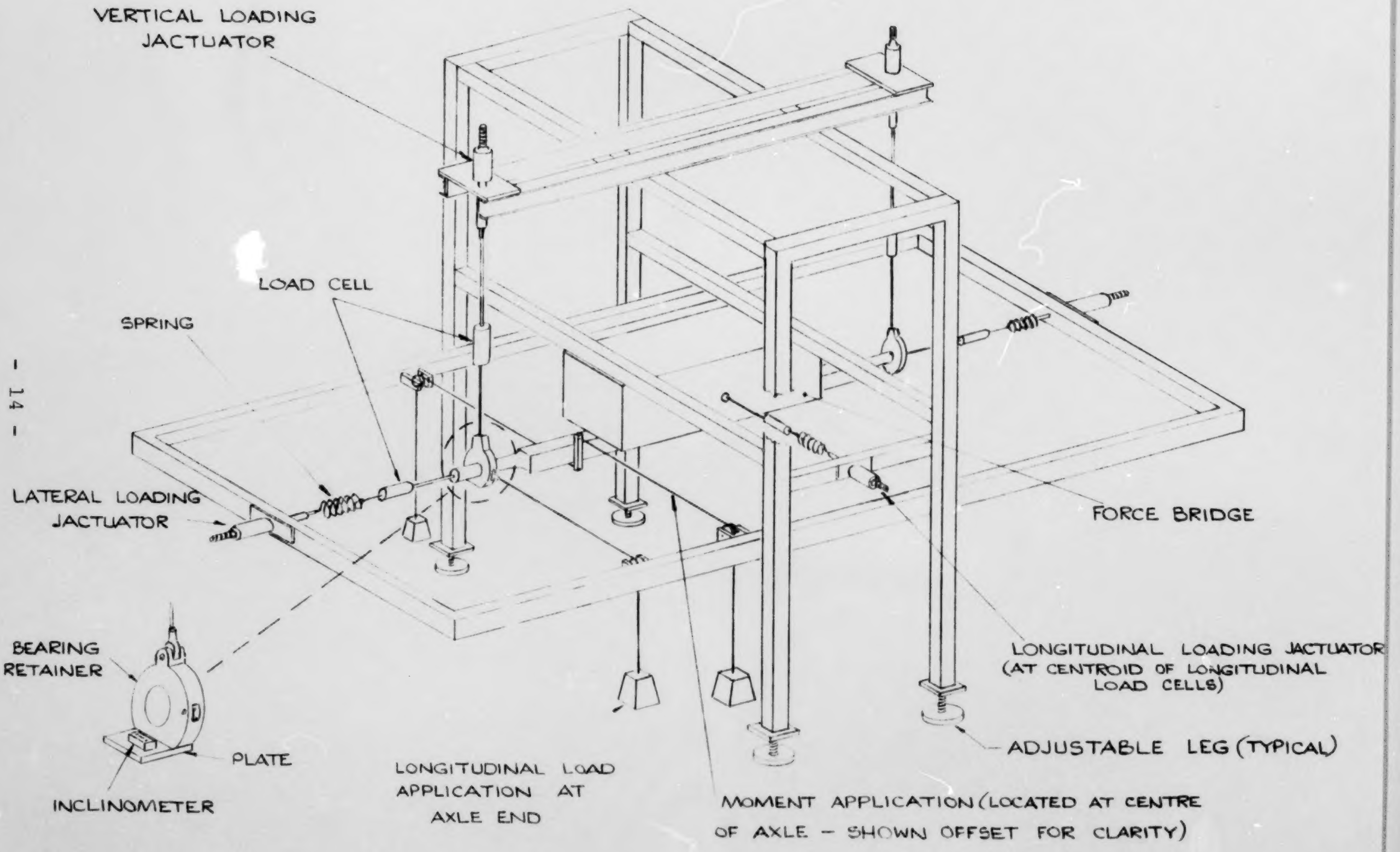

VERTICAL LOADING JACTUATOR
LOAD CELL
SPRING
LATERAL LOADING JACTUATOR
BEARING RETAINER
PLATE
INCLINOMETER
LONGITUDINAL LOAD APPLICATION AT AXLE END
MOMENT APPLICATION (LOCATED AT CENTRE OF AXLE - SHOWN OFFSET FOR CLARITY)
ADJUSTABLE LEG (TYPICAL)
LONGITUDINAL LOADING JACTUATOR (AT CENTROID OF LONGITUDINAL LOAD CELLS)
FORCE BRIDGE

Fig. 4 — SCHEMATIC OF CALIBRATION RIG

3.2.1.2 Calibration Procedures

The generalized theory and methodology of the calibration are given in Appendix A. The application of the theory is described in the following sections.

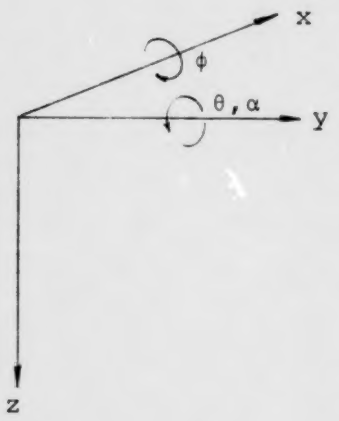
3.2.1.2.1 Force Bridge

The force bridge was calibrated using the rig described in Section 3.2.1.1. The calibration was done in two steps; a linearity check followed by the full calibration.

The purpose of the linearity check was to determine if the force bridge contained any non-linearities or other abnormalities which should be rectified prior to a full calibration. During this check, the Hendrickson suspension was replaced with steel blocks so that no non-linearities would arise from the suspension. In the procedure, the loads in each direction were individually applied and the outputs from the force bridge cells recorded. In all cases where non-linear outputs were observed, they resulted from the load alignment shifting rather than any deficiencies with the force bridge. Refinements in the loading apparatus as well as the procedures were made, as required, to minimize these undesirable effects before the next phase of the calibration.

The full calibration was performed with the Hendrickson suspension in place. With exception to the vertical load, the applied loads were chosen to exceed the forces that the force bridge would experience in service. The force bridge was calibrated for one nominal vertical load. The load ranges and angular excursions that the force bridge was calibrated to, and the co-ordinate system used, are given in Table 1.

Vertical: 8074 Kg (17,800 lbs)
Lateral: ±1361 Kg (3,000 lbs)
Longitudinal: ±227 Kg (500 lbs) - each end
Moment: ±136 N-M (1200 in-lb.)
Pitch Angle θ: ±3°
Roll Angle ϕ: ±3°
Pitch Angle α: ±3°



Axes definitions:

x - towards front of trailer
y - towards r.h.s. of trailer
z - vertically downwards
θ - pitch about 'y' axis, use r.h. rule
α - pitch about 'y' axis, use r.h. rule
ϕ - roll about 'x' axis, use r.h. rule

Table 1 - Load and Angular Rotation Ranges Used for Calibration

The sequence which was used for applying the loads and angular rotation is indicated in Table 2.

Values which were recorded while the various loads or rotations were applied included the following:

- ambient temperature,
- excitation voltage,
- outputs from both loading and force bridge load cells,
- outputs from inclinometer, and
- values of weights used or applicable.

A calibration was also determined for each channel of the data acquisition system, to determine any constants from analog to digital conversion.

3.2.1.2.2 Tangent Angle Rolling Radius (TARR) System

The TARR system was calibrated with the force bridge and TARR system installed on the trailer and with a nominal vertical load of 8074 kg (to compress the suspension). The purpose of this calibration was to determine the outputs of the two LVDT's used in the TARR system when the rolling radius is varied and the angle (α) is varied.

The range and increments that each of the two were varied were as follows:

Rolling radius: range 41 cm (16 in) to 56 cm (22 in),
increments 2.54 cm (1 in)
Angle (α) : range $\pm 3^{\circ}$, increment 1.5°

The general procedure used to vary these inputs was as follows:

a) Set ϕ (roll angle) and θ (pitch angle) to zero and establish a reference rolling radius,

Calibration Case	Applied Loads (Kg)									Angular Rotations (°)			
	Vertical	Lateral		Longitudinal				Moment (N.M)		Range			Increment
				R.H.S.		L.H.S.							
		Range	Increment	Range	Increment	Range	Increment	Range	Increment	θ	α	ϕ	
LATERAL	8074	±1361	454	0	0	0	0	0	0	0	0	0	0
LONGITUDINAL	8074	0	0	± 227 0	113.5 0	0 ± 227	0 113.5	0	0	0	0	0	0
MOMENT	0	0	0	0	0	0	0	136	45	0	0	0	0
PITCH (α)	8074	0	0	0	0	0	0	0	0	0	±3	0	1.5
PITCH (θ) *	0	0	0	0	0	0	0	0	0	±3	0	0	1.5
ROLL (ϕ) *	0	0	0	0	0	0	0	0	0	0	±3	0	1.5

* Performed with and without test wheel/tire assemblies installed.

Table 2 - Calibration Sequence

b) Increment α (pitch angle) by jacking or lowering the front of the trailer,

c) For each increment of α, increment the rolling radius over the total range by placing spacers under the TARR wheels, and

d) Obtain values of α (as measured by the longitudinal inclinometers of the force bridge), and the LVDT outputs for each increment of α and rolling radius.

As in the force bridge calibration, the calibrations were performed at normal room temperature of 21 ±3°C.

3.2.1.3 Data Reduction

The calibration of the force bridge and TARR section is necessary in order to account for the effects of any geometric imperfections in the assembly, preloads and offsets of the load cells in the force bridge, and any cross talk of the sensors. The detailed formulation and methodology for the implementation of the calibration data is given in Appendix A. A brief description of the methodology, the detailed implementations of the methodology and the detailed reduced calibration results are given in the following sections. Plots of the raw calibration data are given in Appendix B.

In brief, the principle of the calibration of the force bridge is as follows. The outputs of the load cells resulting from each applied axle end force or rotational displacement of the force bridge, are the influences that these inputs have on the outputs.

The loads and rotations applied during the calibration are used to determine these influences and any offsets for the normal working range of the bridge.

During the operation of the rolling resistance rig, outputs are obtained from the force bridge load cells, the inclinometers and the TARR LVDT's. By using the influence coefficients obtained in the calibration and correcting for any offsets, the outputs from the rig determine its axle end forces. Because the rotational effects of the force bridge are accounted for, the resulting axle end forces are determined for an equivalent horizontal road surface. From the axle end longitudinal forces, the measured rolling radius of the test tires, and the bearing moment, the total rolling resistance of the four test tires is determined (parallel to the road).

Compared with the methodology described in Appendix A, some aspects of the calibration procedure and reduction of data were simplified for expediency in obtaining results. These included:

- calibration for one nominal vertical load only,
- generally cycling the applied loads only once, and
- excluding the application of the error analysis.

By ignoring those terms which are not affected by the above simplifications, the matrix equation 4.2.1 of Appendix A becomes:

$$[VT_i] - [b_i] - [C\theta_i]\theta - [C\phi_i]\phi - [C\alpha_i - C\theta_i]\alpha = [C_{ij}][F_{oj}] \qquad (1)$$

from which

$$[F_{oj}] = [d_{ji}]\{[V_{Ti}] - [b_i] - [C\theta_i]\theta - [C\phi_i]\phi - [C\ _i - C\theta_i]\alpha\} \qquad (2)$$

where $[d_{ji}] = [C_{ij}]^{-1}$

$[F_{oj}]$ = vector of force bridge axle end forces, $= [F]$

$[C_{ij}]$ = matrix of load cell influence coefficients obtained from calibration $= [C]$

$[V_{Ti}]$ = vector of force bridge load cell outputs $= [V_t]$

$[bi]$ = vector of force bridge load cell zero offsets

$[C\theta_i]$ = vector of influence coefficients due to angular rotation θ

θ = pitch angule rotation of force bridge from horizontal as measured by an inclinometer

$[C\phi_i]$ = vector of influence coefficients due to angular rotation ϕ

ϕ = roll angle rotation of force bridge from horizontal as measured by an inclinometer

$[C\alpha_i]$ = vector of influence coefficient due to angular rotation

α = pitch angle of rotation of force bridge longitudinal axis with respect to the road surface, as measured by the TARR system.

The influence coefficients were determined from the slopes of the data used in the plots given in Appendix B. For accuracy, the least squares method was used to obtain the fits. As additional information, correlation coefficients were determined in each case to assess the goodness of the fits.

It was noted in deriving the above coefficients for $[C]$, that zeros may appear on the diagonal. For that reason (compared with that given in Appendix A), the rows were interchanged so that the resulting vectors appeared in the following order:

$[F]$: F_1, F_2 = axle end force in longitudinal directions
F_3 = axle end forces in lateral directions
F_4, F_5 = axle end forces in vertical directions
F_6 = moment about axle.

$[V_T]$: V_1, V_2 = outputs from lower longitudinal load cells
V_3 = output from lateral load cell
V_4, V_5 = outputs from vertical load cells
V_6 = output from upper longitudinal load cell

With the coefficients substituted in equation (2), as a check on the formulation, the initial trials were attempted by using the voltage outputs obtained during the calibration. In this manner, the output axle end forces could be compared with the known input values. In some cases, considerable discrepancies were observed between the two. An investigation into the causes for these discrepancies lead to two factors, both contained in $[C]$ and consequently its inverse, $[D]$:

i) One coefficient in particular was of very low magnitude and had poor correlation. This term was subsequently set to zero.

ii) $[C]$ was ill behaved. Alternative methods were applied to condition its inverse $[D]$.

With these refinements, the resulting coefficients of equation (2) which were used to reduce data collected in the road trials are in the following.

$$[C] = \begin{bmatrix} \text{3.4500E-04} & \text{-2.0910E-04} & \text{-7.3700E-05} & 0.0000 & 0.0000 & \text{3.5200E-06} \\ \text{-2.3300E-04} & \text{3.7410E-04} & \text{8.2100E-05} & 0.0000 & 0.0000 & \text{4.3400E-06} \\ \text{-3.4100E-06} & \text{3.9350E-06} & \text{-7.5800E-05} & 0.0000 & 0.0000 & \text{-6.9000E-08} \\ \text{1.4800E-06} & \text{-1.8180E-06} & \text{-2.7500E-06} & \text{3.0300E-05} & \text{-5.0600E-05} & \text{6.0000E-08} \\ \text{-1.7900E-06} & \text{1.5900E-06} & \text{3.3300E-06} & \text{-4.4900E-05} & \text{2.6900E-05} & \text{1.6200E-07} \\ 0.0000 & \text{-1.3370E-06} & \text{2.8000E-08} & 0.0000 & 0.0000 & \text{-4.5200E-06} \end{bmatrix}$$

$$[D] = \begin{bmatrix} \text{4.6728E 03} & \text{2.6516E 03} & \text{-1.6691E 03} & 0.0000 & 0.0000 & \text{6.2105E 03} \\ \text{2.9330E 03} & \text{4.3161E 03} & \text{1.8255E 03} & 0.0000 & 0.0000 & \text{6.4005E 03} \\ \text{-5.7166E 01} & \text{1.0594E 02} & \text{-1.3022E 04} & 0.0000 & 0.0000 & \text{2.5599E 02} \\ \text{-1.0885E 02} & \text{-6.4070E-01} & \text{-7.5138E 02} & \text{-1.8464E 04} & \text{-3.4732E 04} & \text{-1.5638E 03} \\ \text{-3.1809E 01} & \text{-8.5171E 01} & \text{1.4265E 02} & \text{-3.0819E 04} & \text{-2.0798E 04} & \text{-1.2633E 03} \\ \text{-8.6792E 02} & \text{-1.2760E 03} & \text{-6.2064E 02} & 0.0000 & 0.0000 & \text{-2.2313E 05} \end{bmatrix}$$

$$[b] = \begin{bmatrix} -.02007 \\ -.02478 \\ -.00196 \\ 0.1456 \\ 0.14937 \\ 0.02141 \end{bmatrix} \quad [C_\theta] = \begin{bmatrix} -.0183 \\ -.01895 \\ 0.0 \\ 0.0 \\ 0.0 \\ -.01533 \end{bmatrix} \quad [C_\phi] = \begin{bmatrix} -.0424 \\ .0473 \\ .0357 \\ .00187 \\ -.001855 \\ 0.0 \end{bmatrix} \quad [C_\alpha] = \begin{bmatrix} 0.12296 \\ 0.11704 \\ 0.0 \\ 0.0 \\ 0.0 \\ -.01179 \end{bmatrix}$$

Rolling resistance (R_R) is obtained from the following formula.

$$R_R = F_1 + F_2 - \frac{F_6}{r} \qquad (3)$$

where r = rolling radius obtained from the outputs of the two LVDT's of the TARR system.

$$= (LVDT_1 + LVDT_2) \times (-3.188) + 19.484 \qquad (4)$$

Equation (4) was determined from the calibration of the TARR system. Values $LVDT_1$ and $LVDT_2$ are the voltage outputs of the two LVDT's of the TARR system. Similarly, the expression for α was determined from the calibration of the TARR system and is as follows:

$$\alpha = (LVDT_1 - LVDT_2) \times 0.92739 - 0.00298. \ (v/v) \qquad (5)$$

3.2.2 Data Acquisition System

A custom, computer-based, mobile data acquisition system and driver display has been built for this test program. The system is used to verify and monitor various inputs prior to a test patch (to confirm stability of test conditions) and is switched to an acquisition mode at the test patch. The system utilizes an LSI-11 microprocessor with bubble memory for data acquisition, and an 8085 microprocessor in the driver display.

Low-level signals from the force bridge are amplified with an electronics package, located within the controlled temperature enclosure. The resultant high-level signals, along with high-level outputs from the TARR sensors and inclinometers and processed signals from the tire internal temperature sensors, are cabled forward at full bandwidth to the tractor cab. An interface in the cab provides physical test points for each signal and connections to overload limit trip circuits, which lower the trailing axle if the load cells/flexures appear endangered.

Also from this interface, connection is made into the data acquisition system proper. Inputs are actively filtered (low pass, 0.083 Hz. cutoff) prior to analog-to-digital conversion (12-bit). While crossing a test patch, the system samples all inputs and stores results in temporary memory; at the conclusion of the patch (which is defined by time, typically 10-15 seconds) results are transferred to non-volatile bubble memory. At the conclusion of a test series, the bubble memory is physically transferred to the in-house computer system for reading and resetting. Dependent on data collection rate and patch length, approximately 20 test patches can be accommodated on one bubble memory card.

The driver/observer display is a compact, dash-mounted, 20 character alphanumeric, fluorescent readout, with a 2-digit thumbwheel switch and pushbutton for control. In the pre-test mode, sensor inputs are available by dialing a two-digit code; the display is scaled and presented with appropriate units. Some inputs and changes to the operation of the data acquisition system are possible via the thumbwheels/pushbutton; e.g., a test number can be entered and the recording time can be altered.

Data acquisition is initiated at the edge of a test patch by pushing the display button; the display indicates "test in progress" over the patch and then reverts to whatever monitoring it was engaged in beforehand. As test results become known, we expect to include some on-board processing of average rolling resistance which will also be available on the display following a test patch.

The data acquisition and data reduction systems are shown schematically in Figure 5.

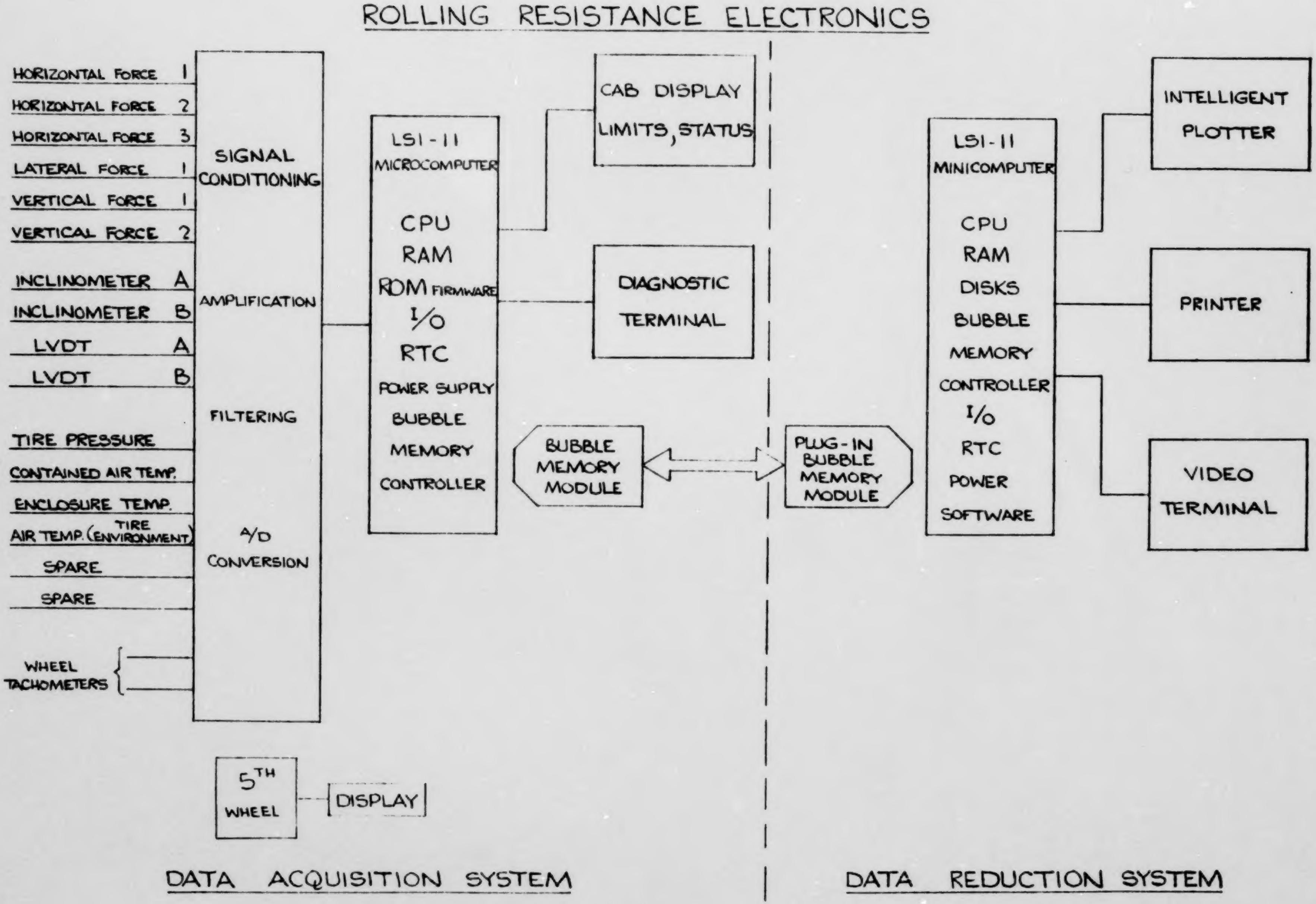

ROLLING RESISTANCE ELECTRONICS
HORIZONTAL FORCE 1
HORIZONTAL FORCE 2
HORIZONTAL FORCE 3
LATERAL FORCE 1
VERTICAL FORCE 1
VERTICAL FORCE 2
INCLINOMETER A
INCLINOMETER B
LVDT A
LVDT B
TIRE PRESSURE
CONTAINED AIR TEMP.
ENCLOSURE TEMP.
TIRE AIR TEMP. (ENVIRONMENT)
SPARE
SPARE
WHEEL TACHOMETERS
SIGNAL CONDITIONING
AMPLIFICATION
FILTERING
A/D CONVERSION
LSI-11 MICROCOMPUTER
CPU
RAM
ROM FIRMWARE
I/O
RTC
POWER SUPPLY
BUBBLE
MEMORY
CONTROLLER
CAB DISPLAY
LIMITS, STATUS
DIAGNOSTIC
TERMINAL
BUBBLE MEMORY MODULE
PLUG-IN BUBBLE MEMORY MODULE
5TH WHEEL
DISPLAY
LSI-11 MINICOMPUTER
CPU
RAM
DISKS
BUBBLE
MEMORY
CONTROLLER
I/O
RTC
POWER
SOFTWARE
INTELLIGENT PLOTTER
PRINTER
VIDEO TERMINAL
DATA ACQUISITION SYSTEM
DATA REDUCTION SYSTEM

Fig. 5 - Rolling Resistance Electronics

3.2.3 Tire Air Pressure and Trailer Rear Axle Lift Controls

The existing tractor trailer air brake system is used to supply air for the test tires, the tag axle air suspension and the TARR system. Each system will be discussed in turn starting with the tire pressure control system. A schematic of the pneumatic systems is in Figure 6.

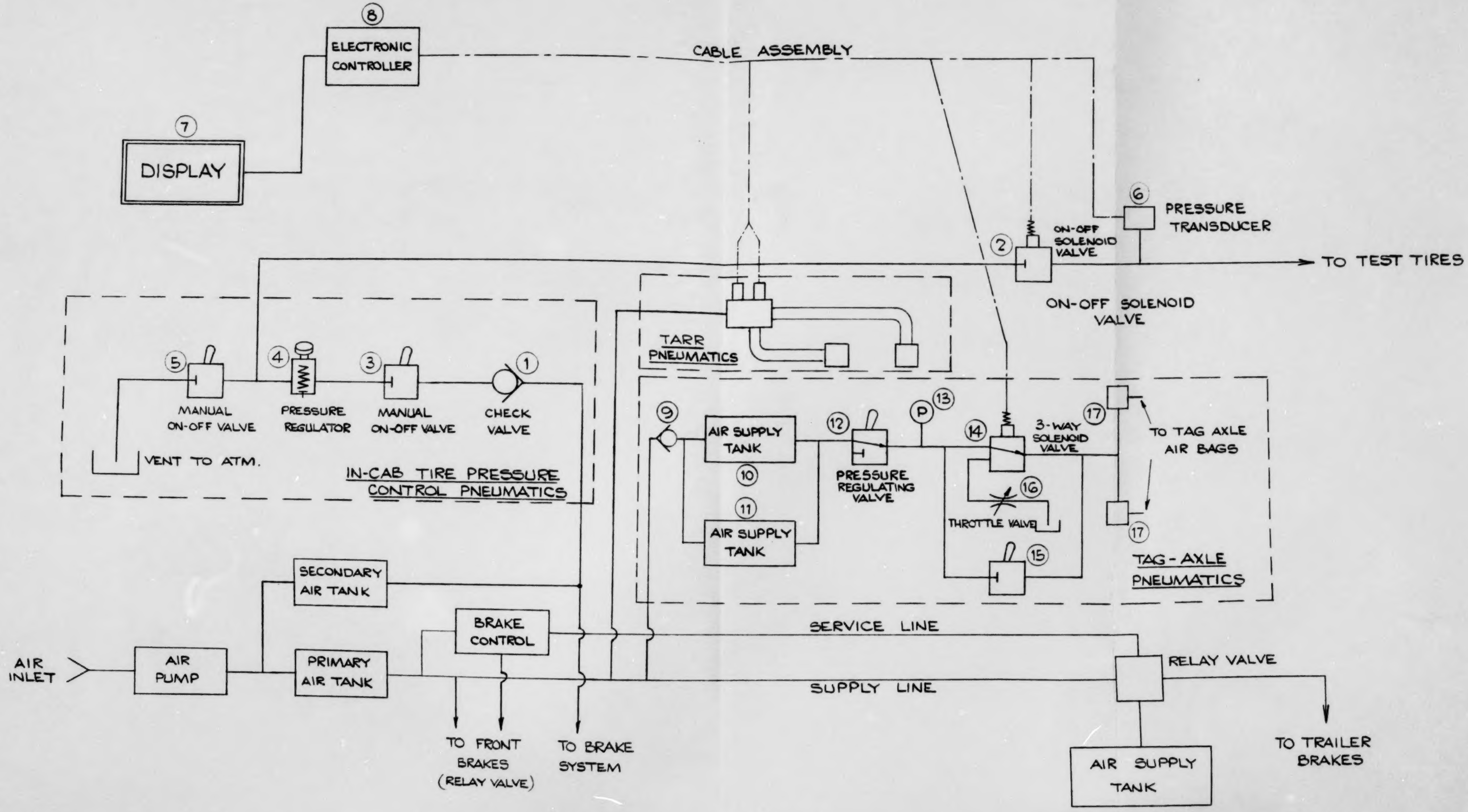
3.2.3.1 Tire Pressure Control System

The air supply for the tires is drawn from the secondary air supply tank of the tractor/trailer since it is only used in the event that the primary tank is partially depleted (i.e. falls below a certain pressure). This secondary tank should have a higher average pressure than the primary tank especially in the event of frequent braking or raising the rear lift axle. Air is transferred to the test tires by way of rotary seals.

Following is a description of the system with major emphasis on each of the control components.

1. Single Check Valve: This valve stops air flow to the tractor brake system in the case of a pressure drop or line failure.

2. On/Off Solenoid Valve: This is a 3-way solenoid valve commonly in tractor trailer air system applications. The valve is a dual solenoid (both normally closed) valve where one leg is from supply to service and the other leg from service to exhaust. Only the supply/service part of the valve is used in this application. The exhaust port is blocked. The main purpose of this valve is to prevent loss of air pressure in the tires should a leak develop in the supply circuit. The valve is closed when the control system senses a pressure loss from



8
ELECTRONIC CONTROLLER
CABLE ASSEMBLY
7
DISPLAY
6
PRESSURE TRANSDUCER
2
ON-OFF SOLENOID VALVE
TO TEST TIRES
ON-OFF SOLENOID VALVE
TARR PNEUMATICS
5
4
3
1
MANUAL ON-OFF VALVE
PRESSURE REGULATOR
MANUAL ON-OFF VALVE
CHECK VALVE
VENT TO ATM.
IN-CAB TIRE PRESSURE CONTROL PNEUMATICS
9
AIR SUPPLY TANK
10
11
AIR SUPPLY TANK
12
PRESSURE REGULATING VALVE
P
13
14
3-WAY SOLENOID VALVE
17
TO TAG AXLE AIR BAGS
16
THROTTLE VALVE
15
17
TAG-AXLE PNEUMATICS
SECONDARY AIR TANK
BRAKE CONTROL
SERVICE LINE
AIR INLET
AIR PUMP
PRIMARY AIR TANK
SUPPLY LINE
RELAY VALVE
TO FRONT BRAKES (RELAY VALVE)
TO BRAKE SYSTEM
AIR SUPPLY TANK
TO TRAILER BRAKES

Fig. 6 — SCHEMATIC OF PNEUMATIC SYSTEM

the pressure transducer in the tire circuit. The solenoid valve is also used in capped pressure tests in which case it will be closed.

3. Manual On/Off Valve: This valve is a normal hand actuated on/off type valve (brass ball valve) (¼" pipe or ½" pipe). It is used for the same function as #2 in case of an electrical failure and is also used when lowering the tire inflation pressure. It is closed while valve #6 is opened.

4. Precision Pressure Regulator Norgren PN 11-018-110: This valve provides a control of .08 - .16 psi at 100 psi inlet with 3 to 5 CFM flow. This regulator is used to set the pressure to the test tires on the trailer and is normally operated by the driver.

   A normal pressure gage was installed for a quick check of the pressure during testing. The digital readout in the cab which receives its signal from the pressure transducer is used for final setting of the pressure.

5. Same as 3. Valve to dump air to atmosphere from the tires. Also used for decreasing tire pressure quickly.

6. Pressure Transducer: Precision pressure transducer used in data acquisition and control system. Two functions:
   a) Monitor & record tire pressure
   b) Signal controller in case of high rate of pressure change or low pressure in system, i.e. leakage detection.

7. Display: electronic digital display to indicate to test driver various parameters including tire pressure.

8. Electronic Controller: Monitors signals from pressure transducer (6) and activates solenoid valves accordingly in case of a failure.

If the pressure at (6) drops below a preset level or drops at a certain rate, solenoid valve (2) would be closed to prevent further leakage of air and/or supply/service part of valve (14) would be opened to inflate tag axle bags. A warning device exists in the cab of the tractor signalling the test driver of a pressure failure.

3.2.3.2 Tag Axle Control

The tag axle suspension received its air supply from the trailer brake system supply line. A tee was placed in the supply line for this purpose.

9. Single Check Valve (existing): prevents air flow from air tanks (10) and (11) to brakes air supply line.

10. Air Tank (existing): to supply air for tag axle air bags.

11. Air Tank (installed): for extra air supply in order to decrease time required to drop the tag axle.

12. Pressure Regulating (on/off/exhaust) valve (existing): Presently used to raise and lower tag axle. In the future could be used only to regulate pressure to tag axle suspension.

13. Pressure Gage (existing): to monitor air pressure to tag axle air bags.

14. 3-Way Solenoid Valve: This valve is similar to (2) and is normally closed. This valve is used to normally raise and lower the tag axle from inside the cab via a toggle switch.

15. Manual Bypass Valve: This is used while transporting the trailer long distance on the tag axle. The valve bypasses

the solenoid valve so that the tag axle remains down in the event of an electrical failure.

16. Throttling Valve: This valve is set to control the rate of pressure drop when the quick release valves (17) exhaust the air from the tag axle suspension in order to minimize the impact load on the test axle when it contacts the road. The valve is closed while valve (15) is activated.

17. Quick Release Valves (existing): These valves are used to exhaust air from the air bags.

3.2.3.3 TARR Actuation

The TARR wheels are raised or lowered via a toggle switch in the tractor cab which activates two solenoid controlled dual-acting air cylinders.

3.2.4 Tire Contained Air Temperature Measurement System

The tire contained air temperature is sensed by a silicon temperature probe located at the approximate center of the cavity of one tire. The inner tube and rim were modified and sealed to accept the probe. Signals from the probe are transmitted to the data acquisition system through a custom designed slip ring assembly attached to the wheel hub. The slip ring assembly contains extra rings in the event that the use of additional probes may be denied at a later date.

3.2.5 Constant Temperature Enclosure Requirements

A preliminary evaluation of the requirements was made for the force bridge constant temperature enclosure. It was assured that the test trailer would normally be stored at room temperature conditions (20° C). Therefore, the heating or cooling requirements would be the amounts required to maintain the soak temperature while the test trailer is in use at other temperatures. Considering conduction, convection and losses due to leakage, the total estimated heat loss is 192 BTU/hr/°F.

Assuming that the rolling resistance rig is operated in a temperature range of -10°F to +90°F, the approximate heating and cooling requirements are as follows:

Heating: 15,400 BTU/hr.
Cooling: 3,850 BTU/hr.

3.2.6 Road Surface Measurement System

A hand held probe, Fluke model 80T-150, was used for contact measurements with the road surface. The probe is a small semiconductor element at the end of a pen-like probe, and provides digital readings in degrees Celsius over the range -50 to +150 degrees. Accuracy from 0 to 100 degrees is ±1 degree.

3.2.7 Fifth Wheel

A Nucleus fifth wheel with optical tachometer and digital speed display (retentive speedometer) was used to provide a calibrated display for the driver.

3.2.8 Air Temperature Measurement System

Air temperature at the test tire location was measured using Analog Devices model AD590 silicon sensors; these convert temperature to current (1 microamp/degree Kelvin), which is measured across a resistor and bussed forward to the data acquisition system.

3.2.9 Error Analysis

The theory and the mathematical derivation for the system error analysis is contained in Appendix A.

3.2.10 Acquisition and Baseline Testing of Tires

Three sets (four each) of tires and suitable rims were acquired for the project. The tires acquired were as follows:

Michelin radial ply x ZA size 11R24.5/G
Goodyear bias ply 10 x 20/F
Goodyear radial ply 11R22.5/G.

The Michelin tires were donated by Michelin Tires of Canada, Ltd.

Following their acquisition, the test tires were tested by the Calspan Corporation in their TIRF facility to obtain laboratory values of rolling resistance. The procedure used was that recommended by SAE Sub-Committee on Rolling Resistance. A summary of these results, taken from (2), are reproduced in Tables 3 and 4.

3.2.11 Safety Systems

A separate system designed to automatically control lowering of the trailer tag axle (and hence unload the measurement bridge) was designed to give a measure of protection against load cell overloads or loss of tire pressure. The system works through a separate indicator/control console located within the driver's field of view in the tractor cab.

TABLE 3

Steady State Rolling Resistance Test Results
for the Three Truck Tire Types (Reference 2)

Run No.	Tire Type	TIRF Tire No.	Steady State Rolling Resistance Force, FR				
			100% Load 100% Press.* (lb)	100% Load 90% Press. (lb)	75% Load 70% Press. (lb)	50% Load 120% Press. (lb)	25% Load 70% Press. (lb)
2-20-2	10 x 20/F	220-9	34.92	-	-	-	-
5-20-2	↓	220-10	35.30	43.65	34.98	15.53	10.01
8-20-2		220-11	34.16	-	-	-	-
11-20-2		220-12	32.30	-	-	-	-
1-20-2	11R22.5/G	220-1	25.34	-	-	-	-
4-20-2	↓	220-2	26.46	29.78	24.85	13.67	8.88
7-20-2		220-3	25.85	-	-	-	-
10-20-2		220-4	25.36	-	-	-	-
13-20-2		220-3	25.96	-	-	-	-
3-20-2	11R24.5/G	220-5	29.83	-	-	-	-
6-20-2	↓	220-6	30.72	34.32	28.85	15.41	9.39
9-20-2		220-7	29.67	-	-	-	-
12-20-2		220-8	29.14	-	-	-	-

*Capped inflation pressure (all other pressure regulated)

NOTE: See Table 4 for actual loads and inflation pressures.

TABLE 4

Multi-Linear Regression Coefficients and the Standard Estimate of Error for the Equation for the form: (See Reference 2)

$$FR = k_0 + k_1 L + k_2 L^2 p^{-1} + k_3 L p^{-1}$$

TIRF Test No.	Tire Type	Test Condition	L Load (lbs)	P Pressure (psi)	FR (lbs)	Resulting Coefficients and Estimate of error
4-20-2	11R22.5/G	1	5268	107.1	26.46	$k_0 = 1.687$
		2	5270	85.7	29.78	$k_1 = .264 \times 10^{-2}$
		3	3959	66.6	24.85	$k_2 = .918 \times 10^{-5}$
		4	2649	114.3	13.67	$k_3 = .178$
		5	1341	66.7	8.88	$e = .510$
5-20-2	10 x 20/F	1	4739	92.4	35.30	$k_0 = 2.816$
		2	4734	67.8	43.65	$k_1 = .205 \times 10^{-2}$
		3	3553	52.7	34.98	$k_2 = .641 \times 10^{-4}$
		4	2376	90.3	15.53	$k_3 = .141$
		5	1178	52.7	10.01	$e = .168$
6-20-2	11R24.5/G	1	5619	107.5	30.72	$k_0 = -.370$
		2	5623	85.8	34.32	$k_1 = .342 \times 10^{-2}$
		3	4228	66.8	28.85	$k_2 = -.116 \times 10^{-5}$
		4	2827	114.5	15.40	$k_3 = .239$
		5	1425	66.8	9.39	$e = .493$

For protection against load cell and flexure overloads, force signals are tapped at the data acquisition system prior to the low-pass (0.083 Hz) filters and fed to absolute-value (bipolar) comparator circuits (six in total). Each comparator circuit has two trip points; a lower level (e.g., 50% of full load) to serve as a caution, and a higher level (e.g., 90% of full scale) to trigger automatic lowering of the tag axle.

Because tire pressure can be externally supplied and varied, loss of pressure through equipment malfunction or tire failure is a possibility and may be more damaging than a normal trailer tire failure. In the event tire pressure is seen to exceed a lower limit, or if it is varying too quickly, the tag axle is tripped.

Tire pressure is sensed electrically and the signal routed to the safety system, where it is amplified and compared to an adjustable lower limit. Rate-of-change of tire pressure is also monitored with a dp/dt circuit which trips the tag axle if pressure is falling faster than the normal bleed-off experienced when adjusting tires during experiments.

The driver's console monitors conditions and permits manual control of the tag axle. Each load cell has an associated indicator which lights at the lower (cautionary) force level, and the driver is expected to take action (e.g., slow down) if possible. If an upper level is exceeded on any load cell, the tag axle is dropped and a "limit exceeded" lamp latched on. Similarly, pressure drop is indicated by a lamp and also trips the tag axle relay and "limit exceeded" lamp.

Following an automatic tag axle actuation, the safety systems must be reset by the driver pushing a button. This cancels the "limit exceeded" lamp and raises the axle.

The console also has tag axle status lamps (down and up) and a manual switch for holding the axle in the lowered position, used for example while driving between test patches.

3.2.12 Road Trials

A short road test program was undertaken with the Goodyear 10:00 x 20 Bias ply tires with nominal load of 18,770 lbs. The tests were conducted on approximately a 20-mile section of Highway 417 (four land paved highway) towards Montreal from Ottawa. It was noted that this section of the highway contained a resurfaced portion, as well as the original surface. Hence, at least two road surface types were tested although records were not kept of the runs which were conducted on each type.

After a warm-up period of approximately 1½ hours at 80 km/hr, test data were obtained (at no specific location of the test section) for speeds varying from 80 km/hr to 20 km/hr with the tires capped (from a cold pressure of 75 psig) and regulated at pressures of 75,65,58 and 52.5 psig. Data samples were taken for durations of approximately 4 seconds to 14 seconds. The results of these tests are given in Table 5 and in Figures 7 to 11. The values of rolling radius, rolling resistance (force and coefficient) and bearing moment are the mean of the values for the test time durations indicated in Table 5.

In Figure 7, the results indicate a general trend of increase in rolling resistance with increase in speed. Figure 8 indicates a general decrease in rolling resistance with pressure increase. Also shown in Figure 8 for comparison, is a plot of the empirical equation based on the results obtained by Calspan for these tires at 80 km/hr. Variation of wheel bearing moment for different speeds and tire inflation pressures are given in Figure 9. Time histories of rolling resistance and bearing moment for a specific test patch are given in Figures 10 and 11, respectively.

Although the results generally are of an expected magnitude (with the possible exception of bearing moment) and show trends previously established by laboratories, there is considerable scatter in the results. This occurred from several possible sources:

a) The test data was taken (more or less) at random sections of the road, hence the data can not strictly speaking be compared.

b) The lateral load (causing steering slip) varied for the various test cases; no corrections were made to normalize the results of rolling resistance or bearing moment.

c) Due to the procedures used in obtaining the test data, in general the tire temperatures were not stabalized, particularly at the lower speeds. Typically the trailer was towed at 80 km/hr, except when data at the lower speeds was being taken.

d) The total sample time was generally quite small.

e) As is evidenced by the scatter in the time history data, particularly Figure 11, the values changed more rapidly with time than they should have, considering that all of the relevant sensors contained a .083 Hz filter. The data indicates that one or more filters were acting intermittently (poor electrical connection). The result of this output, combined with a small sample size, would lead to considerable scatter in the final averaged data.

f) Compared with laboratory results (in which it is assumed low friction wheel bearings would be used), the on-road tests were carried out with perhaps greater braking torque present, the amount varying as indicated by the values of bearing moment in Table 5.

g) Some error would have resulted from temperature drift of the components as the rig was calibrated at nominal temperature of 20°C, while the tests were carried out at an ambient air temperature of 10°C.

h) Further refinements in the calibration data modelling and data reduction could improve the results.

i) The bearing moment results, as reviewed by the Steering Committee, are of a higher magnitude than expected. The possible cause for this requires further investigation.

Several problems were noted prior to, during, or after the road trials were conducted.

a) The bubble memory of the data acquisition system was inadvertently damaged and the replacement component did not arrive in time for the scheduled road trials. As a result, the road trial data required manual input for data reduction which was time consuming and tedious. This was also one reason why relatively short data samples were taken during the trials.

b) The tire contained air temperature measurement was not operating satisfactorily at high speeds. The problem is assessed to be mechanical in nature relating to the slip ring assembly.

c) The inflation pressures were found to be in error from the data acquisition system (voltages were also taken directly and pressures directly from the pressure gage for additional checks). The scaling of the transducer amplification requires checking.

d) The fifth wheel readout was intermittent. The source of the problem is likely due to the fifth wheel leaving the road surface because it is mounted at the rear of the test trailer. Remounting the fifth wheel at the center of the trailer would likely improve this situation.

Test No	Data Sample (sec.)	Test Condition	Speed (Km/h)	Inflation Pressure (psig)	Temperatures (°C)			Average Rolling Radius (in.)	Average Rolling Resistance (lbs)	Average Bearing Moment (in-lb)	Average Rolling Resistance Coef. $(\times 10^{-3})$	Lateral Load (lbs)
					CAT	Road Surface	Ambient Air					
1	5.75	Capped at 75 psig (cold)	80	83.7	59	10	7-12	20.60	180	525	9.6	343
2	2.75		60	83.7	59			20.54	143	653	7.6	167
3	2.75		40	82.3	59			20.47	134	547	7.1	320
4	6.75		20	83.7	63			20.64	134	505	7.1	270
5	5.75	Regulated at 75 psig.	80	75.0	59			20.45	179	578	9.5	288
6	4.75		60		59			20.50	165	573	8.8	374
7	3.75		40		59			20.33	138	602	7.4	293
8	4.75		20		55			20.50	140	548	7.5	456
9	12.75	Regulated at 65 psig	80	65.0	72			20.38	180	594	9.6	352
10	4.75		60		72			20.27	163	578	8.7	315
11	3.75		40		72			20.36	161	524	8.6	264
12	2.76		20		66			20.43	162	328	8.6	340
13	12.75	Regulated at 58 psig	80	58.0	74			20.11	185	627	9.9	329
14	4.75		60		74			20.14	179	630	9.5	325
15	3.75		40		71			19.99	156	592	8.3	270
16	4.75		20		66			20.04	161	430	8.6	352
17	12.75	Regulated at 52.5 psig	80	52.5	85			20.06	198	546	10.6	288
18	4.75		60		77			20.01	191	528	10.2	345
19	3.75		40		74			19.97	183	472	9.8	237
20	3.75		20		74			19.79	192	340	10.2	372

TABLE 5 - TEST RESULTS FOR GOODYEAR 10:00 x 20 BIAS PLY TIRES - VERTICAL LOAD = 18770 lbs.

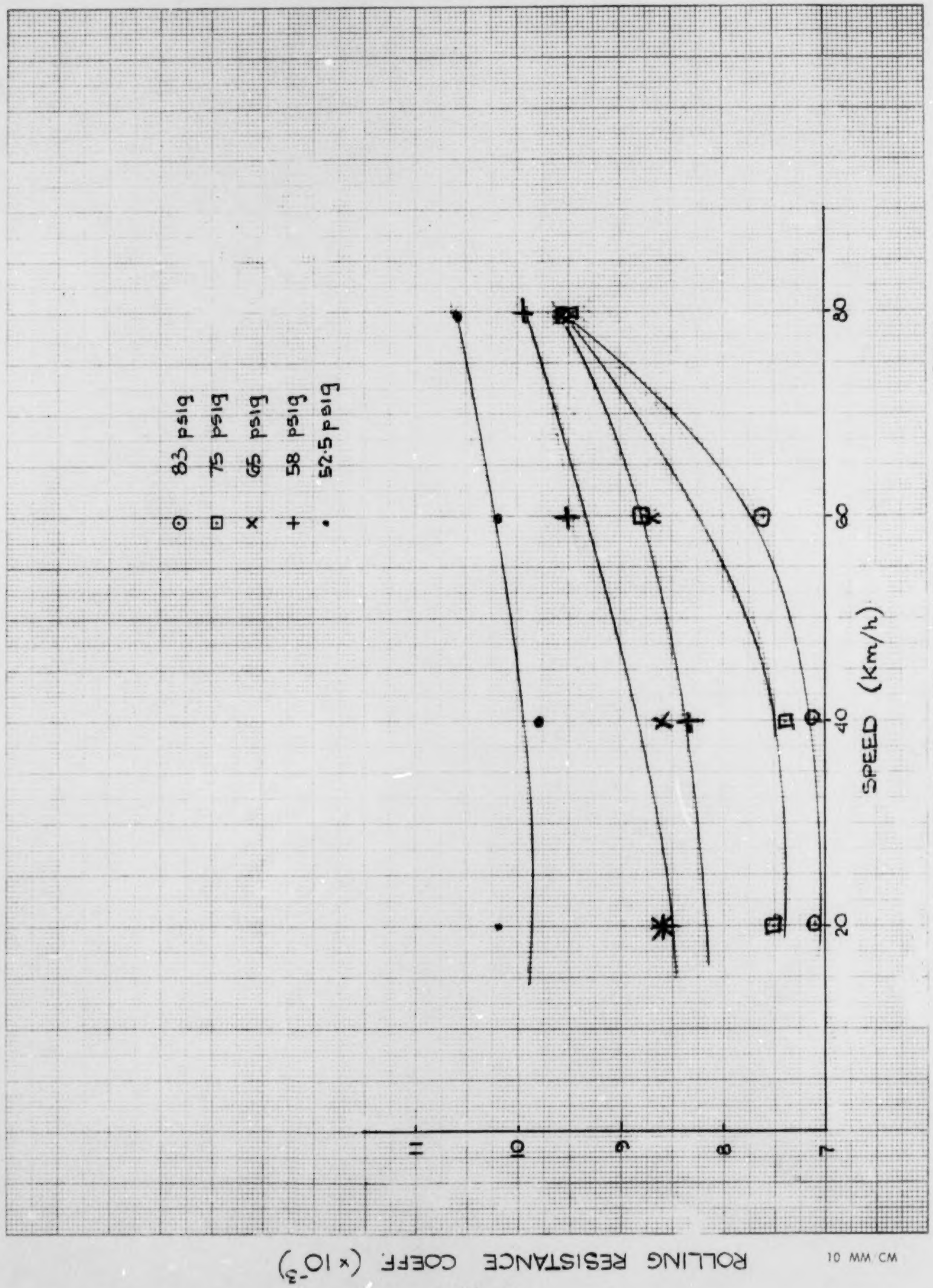


Fig. 7 ROLLING RESISTANCE COEFFICIENT vs SPEED

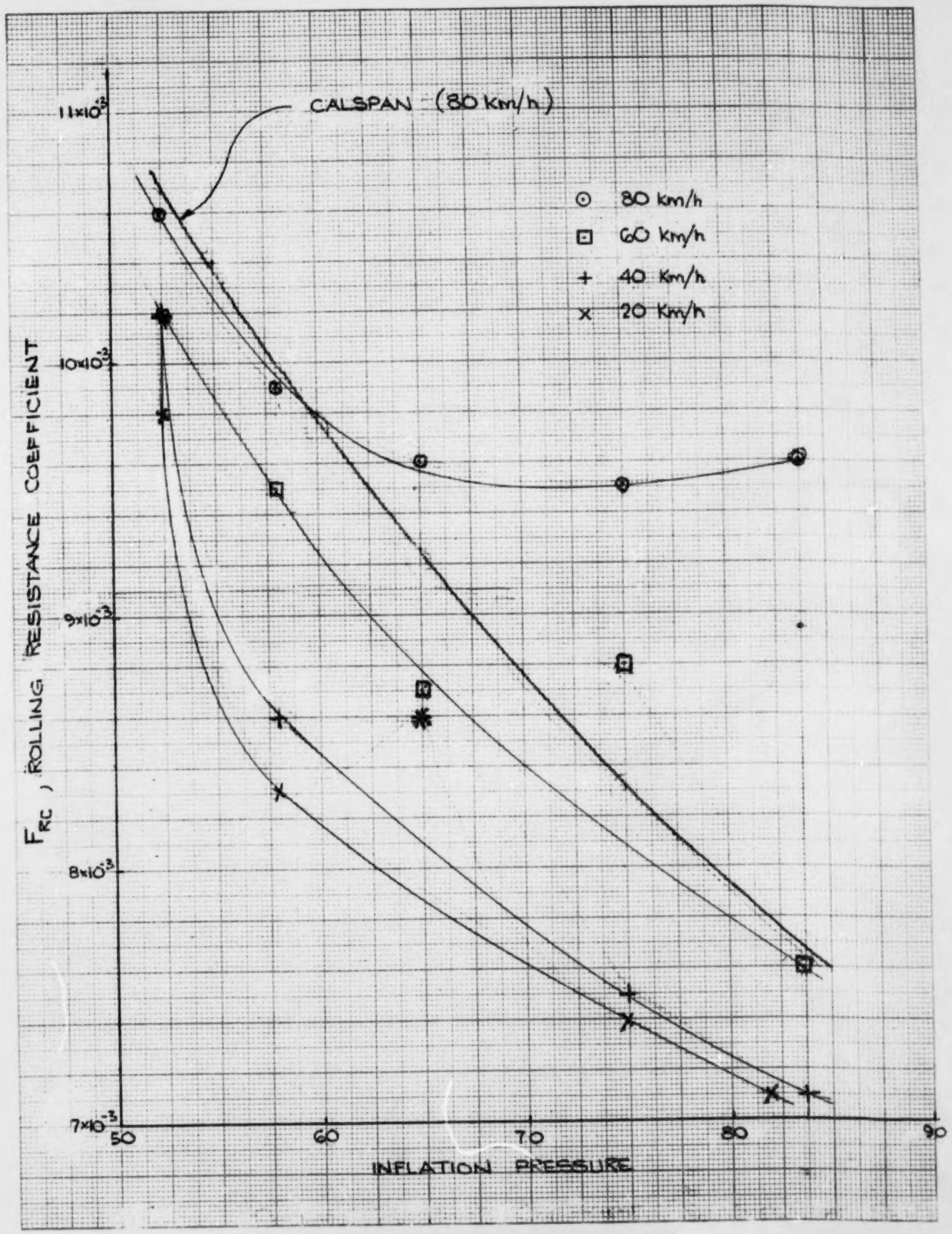


Fig. 8 ROLLING RESISTANCE COEFFICIENT vs INFLATION PRESSURE

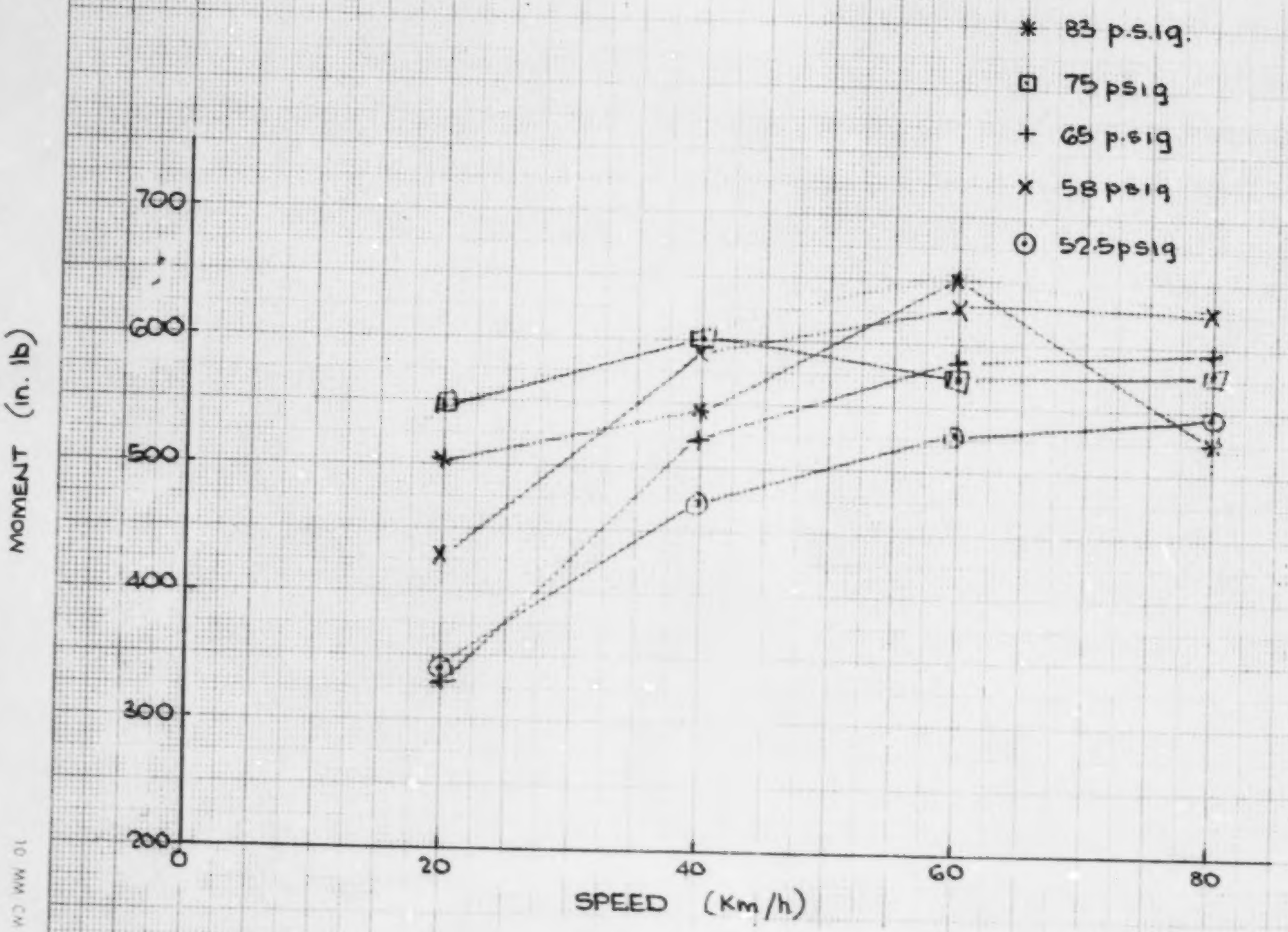


Fig. 9 MOMENT VS SPEED

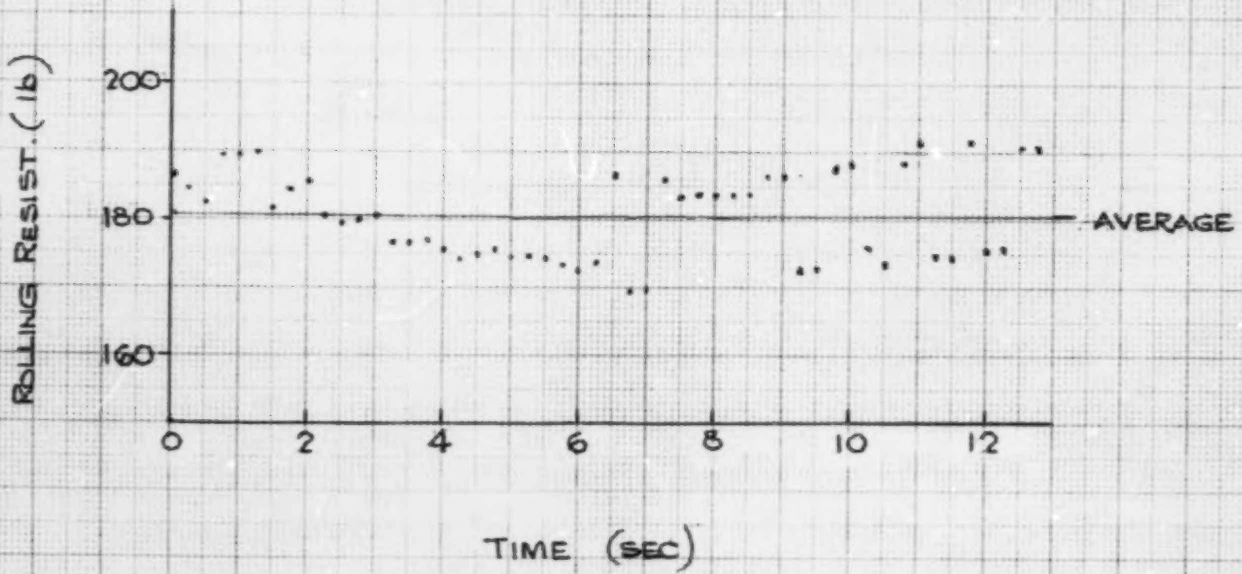


Fig 10 Rolling Resist. vs Time (Pressure = 65 psig, Speed 80 km/h)

10 MM/CM

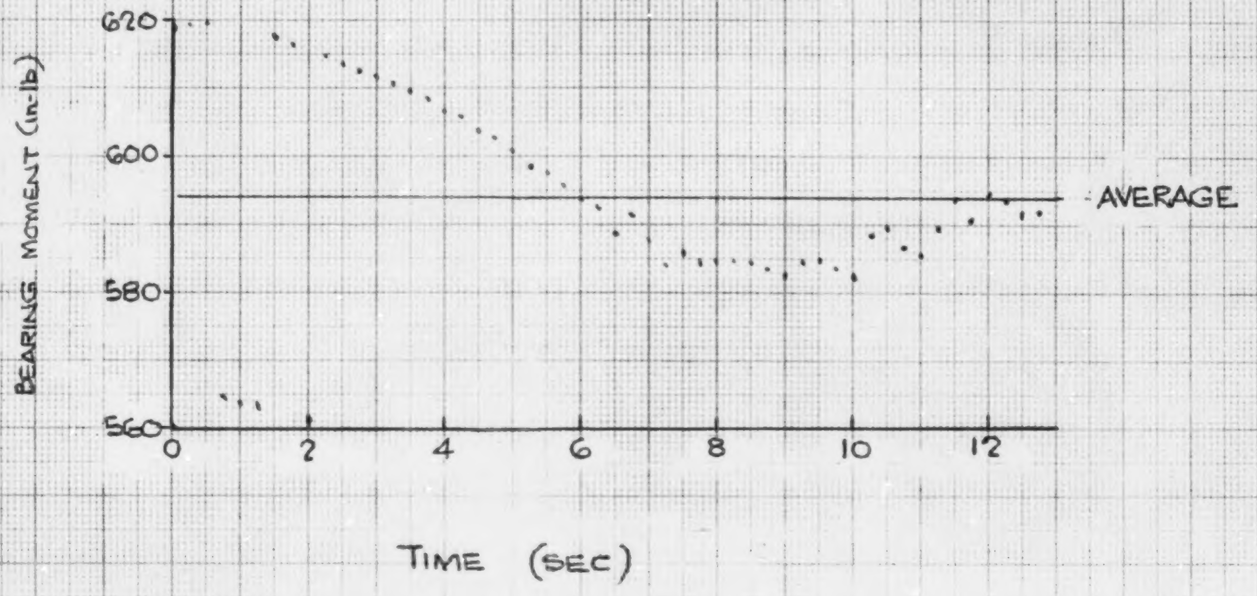


Fig. 11 BEARING MOMENT vs TIME (PRESSURE = 65 psig, SPEED 80 km/h)

10 MM/CM

4.0 CONCLUSIONS AND RECOMMENDATIONS

Significant progress has been made in further developing the rolling resistance rig for measuring over-the-road rolling resistance and bearing moment losses of large truck tires. Although numerous unforeseen problem areas (as can be expected in an R & D project of this nature) were encountered in the course of conducting the project, valuable experience was attained in resolving these problems which can be applied to the further development and refinement of the rig. The limited over-the-road test trials demonstrated that the overall concept was viable and provided valuable information towards the specific areas in which further efforts should be directed. Based on the overall results of the project activities to date, there is optimism that the overall project activities will be met through further efforts.

On the basis of the above, in conjunction with the comments and recommendations made by the project Steering Committee, the following activities are recommended for a future phase of the project:

4.1 - Additional over-the-road testing should be performed to verify the results under equilibrium conditions with the force bridge as presently calibrated. For these tests, the bubble memory should be installed and with the other relatively minor problems (temperature and pressure measurements, signal filtering and fifth wheel) resolved. This activity should be conducted before calibrations for other loads are undertaken.

4.2 - Potential discrepancies in the bearing moment values should be investigated.

4.3 - The temperature sensitivity of the rig should be checked before proceeding with the completion of the constant temperature enclosure. If required, the enclosure should be completed.

4.4 - A method for measuring the trailer slip angle should be investigated since this could have a large effect on the rolling resistance results.

4.5 - The lateral loads obtained over-the-road should be checked against laboratory results.

4.6 - The road test sections should be analyzed.

4.7 - Possible refinements in the data reduction, data acquisition systems and safety systems should be investigated.

4.8 - A static calibration of the force bridge over its full operating range should be carried out.

4.9 - A system error analysis should be carried out.

4.10 - A full test program for the three sets of tires should be carried out for a wide range of operating, ambient temperature and road conditions.

4.11 - If warranted, (depending on wear), laboratory rebaselining of the test tires should be performed.

4.12 - An assessment should be made of the energy penalties associated for different tire/road surface interactions.

4.13 - An assessment should be made of bearing designs and their lubricants on energy losses.

4.14 - Consideration should be given to investigating tire interactions with pavement in the transient stages of warmup and the effects of wheel slip and phase relationships.

5.0 REFERENCES

1. Transport Canada, "Evaluation of Truck Tire Rolling Resistance, Bearing Losses and Road Surface Effects on Rolling Resistance - Phase I", TP 2348, July, 1979.

2. Calspan Corporation,"Results of Rolling Resistance Tests on Twelve Truck Tires Conducted For Davis Engineering", Calspan Report No. 6708-T, June, 1980.

APPENDIX A

Calibration & Error Analysis Theory

1.0 NOTATION

1.1 Axes

The vehicle axis system is detailed in Figure 1.1-1.

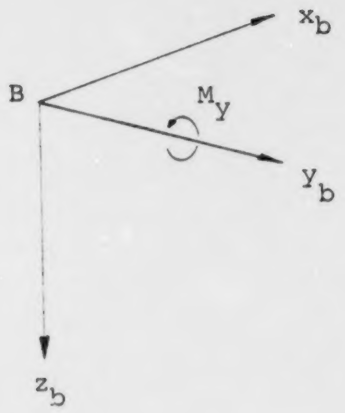


Figure 1.1-1: Vehicle Axis System

Forces and moments applied externally to the bogie are positive in the positive axis convention.

The x_b-axis is aligned with the longitudinal axis of the bogie, positive towards the front of the truck. A rolling resistance force will therefore be negative.

When facing toward the front of the truck, the y_b-axis is positive to the right. Any force that tends to pull the bogie toward the shoulder of the road is therefore positive.

The z_b-axis is positive downward. The y_b-z_b plane defines the vertical plane of the bogie. This plane is not necessarily plumb. The trailer applies a positive load to the bogie but the road surface applies a negative load, equal and opposite. Thus, the vertical loads applied at the axles through bearings are negative.

Moments about the y_b-axis follow the right-hand rule. Therefore, external moments applied in the sense of the bearing moments are negative.

Angle conventions are consistent with the right-hand axis system and are discussed later.

1.2 Measured Outputs

The bogie contains six load cells. Each of these has an output in the form of mV. The output is amplified and converted to a digital readout. Care must be taken to ensure polarities of the supply voltage to each load cell are never reversed. Calibration should be from externally applied loads to A/D output. Because a failure of an A/D would mean loss of calibration data, the mV outputs of every load cell should be recorded as well as the A/D output. If an A/D must be replaced, it can be calibrated separately before installation.

Outputs from the A/Ds are defined as follows:

$V_T(1)$ = output from lower right-hand longitudinal load cell

$V_T(2)$ = output from upper longitudinal load cell

$V_T(3)$ = output from lower left-hand longitudinal load cell

$V_T(4)$ = output from lateral load cell

$V_T(5)$ = output from right-hand vertical load cell

$V_T(6)$ = output from left-hand vertical load cell

1.3 Calibration Loads

Calibration loads are the loads applied external to the bogie, with signs following the convention of Section 1.1.

These are generally more accurate than the precision error associated with the measured outputs so initially at least, errors associated with applied loads will be ignored.

${}^{F}C_{jk}$ is used to denote a calibration load where the first subscript denotes the location of the calibration force and the second subscript identifies the magnitude. Locations are defined as:

${}^{F}C_{1,K}$ = applied load in longitudinal direction at right-hand end of axle

${}^{F}C_{2,K}$ = applied load in longitudinal direction at left-hand end of axle

${}^{F}C_{3,K}$ = applied load in lateral direction through axle centroid

${}^{F}C_{4,K}$ = applied load in vertical direction at right-hand end of axle

${}^{F}C_{5,K}$ = applied load in vertical direction at left-hand end of axle

${}^{F}C_{6,K}$ = applied moment about axle

${}^{F}C_{6,K}$ is actually a moment but the ${}^{F}C$ notation is preserved for consistency.

2.0 FORCE CALIBRATION OF THE BOGIE

2.1 Nominal Vertical Loads

The bogie is calibrated approximately in the middle of its working range to enhance the accuracy of the calibration. The vertical loads are the only ones that differ in their nominal condition appreciably from their zero readings.

A typical nominal load is selected, say 15,000#, as being midway in the working range of 10,000 to 20,000#. Using the calibration curves for the two load cells, the jactuators are adjusted to obtain a mV output for each load cell corresponding to 7,500# and this is maintained throughout the calibration.

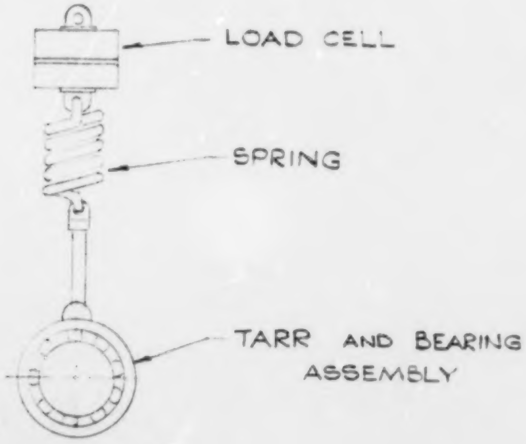


Figure 2.1-1

As illustrated in Figure 2.2-1, a tare weight correction must be applied to all vertical loads to account for the weight of the spring and bearing assembly. This may be done at the end of the force calibration of the bogie prior to angle calibration.

Tare correction consists of hanging the spring and bearing assembly from the load cell and noting the offset in mV. This is then subtracted from all previous readings, i.e., the nominal load on each side may now become 7,380#. Tare correction must be applied to each vertical load cell.

2.2 Intercept Calculation

This section is not intended to identify the individual steps in a calibration run; rather, it is meant to define the required data. From this section, the engineer in charge of calibration can produce the necessary tables for recording data and develop the best procedure for obtaining the data.

Consider, for example, a calibration cycle where the external load in the x_b-direction on the right-hand axle is cycled. All other external loads are zero save for the vertical loads which are at their nominal values.

Every load cell output on the bogie may be expected to cycle to some degree with the applied external load. This cycling will include non-linearity and hysteresis but the slope of a best fit straight line through the data represents the partial differentials of the measured outputs with respect to the applied force, i.e.,

$$\frac{\partial V_{T(1)}}{\partial F_{C_1}} , \frac{\partial V_{T(2)}}{\partial F_{C_1}} \quad \ldots\ldots \quad \frac{\partial V_{T(6)}}{\partial F_{C_1}}$$

Associated with the linearization is a precision error derived from the deviation between each measured data point and the best fit straight line. The presence of the nominal load may result in permanent shift of the ordinate intercept of the remaining four outputs, and to constrain the best fit line to pass through the origin would needlessly degrade the accuracy of the calibration data. Therefore, in the data reduction process, it is necessary to preserve this offset.

As a minimum, each calibration cycle must include nine (9) points beginning from the unloaded condition (or the nominal load condition in the case of the vertical load cells) and ending up with the zero (or nominal) condition. These cycles should be repeated as many times as can be accommodated within the schedule, and in any event, at least two cycles should be required.

The process is repeated for each of the remaining five external loads (including $^{F}C_6 \equiv M_y$) in turn. Errors associated with the applied loads should be estimated and recorded for future reference.

Power supplies for all load cells and A/Ds should be monitored and recorded throughout the process.

Load cell outputs should be recorded in addition to A/D outputs for the reasons discussed in Section 1.2.

The complete family of curves is depicted in Figure 2.2-2 and Table 2.2-1 summarizes the end to end data record for one calibration cycle.

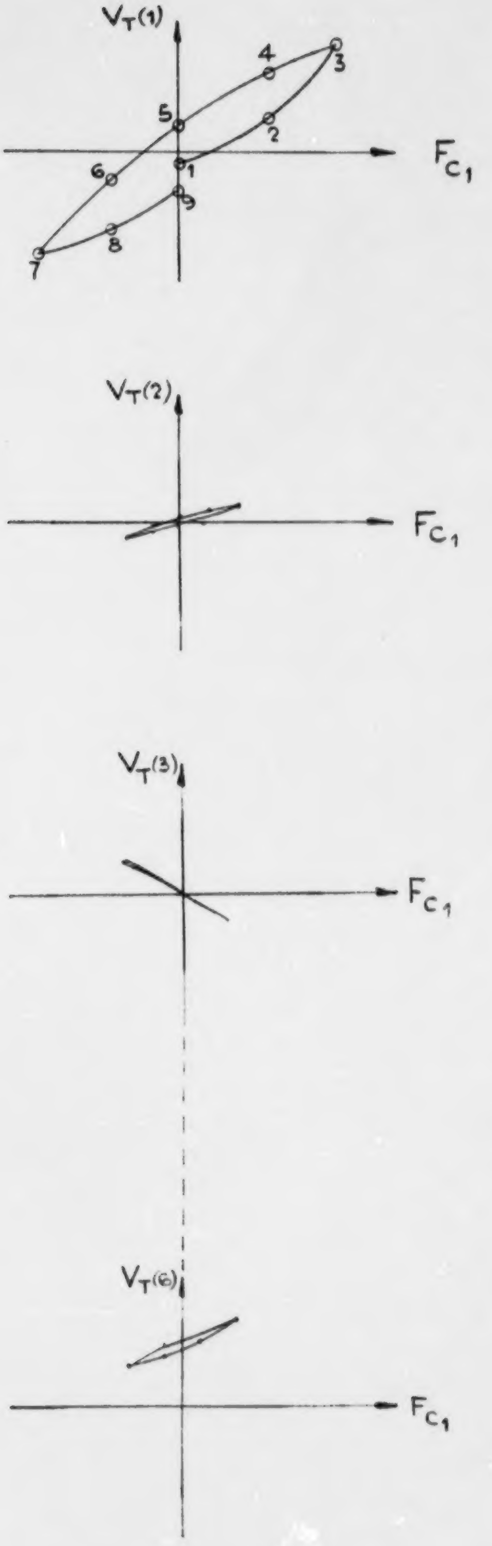

$V_T(1)$
4
3
5
2
F_{C_1}
6
1
9
8
7
$V_T(2)$
F_{C_1}
$V_T(3)$
F_{C_1}
$V_T(6)$
F_{C_1}

Figure 2.2-1

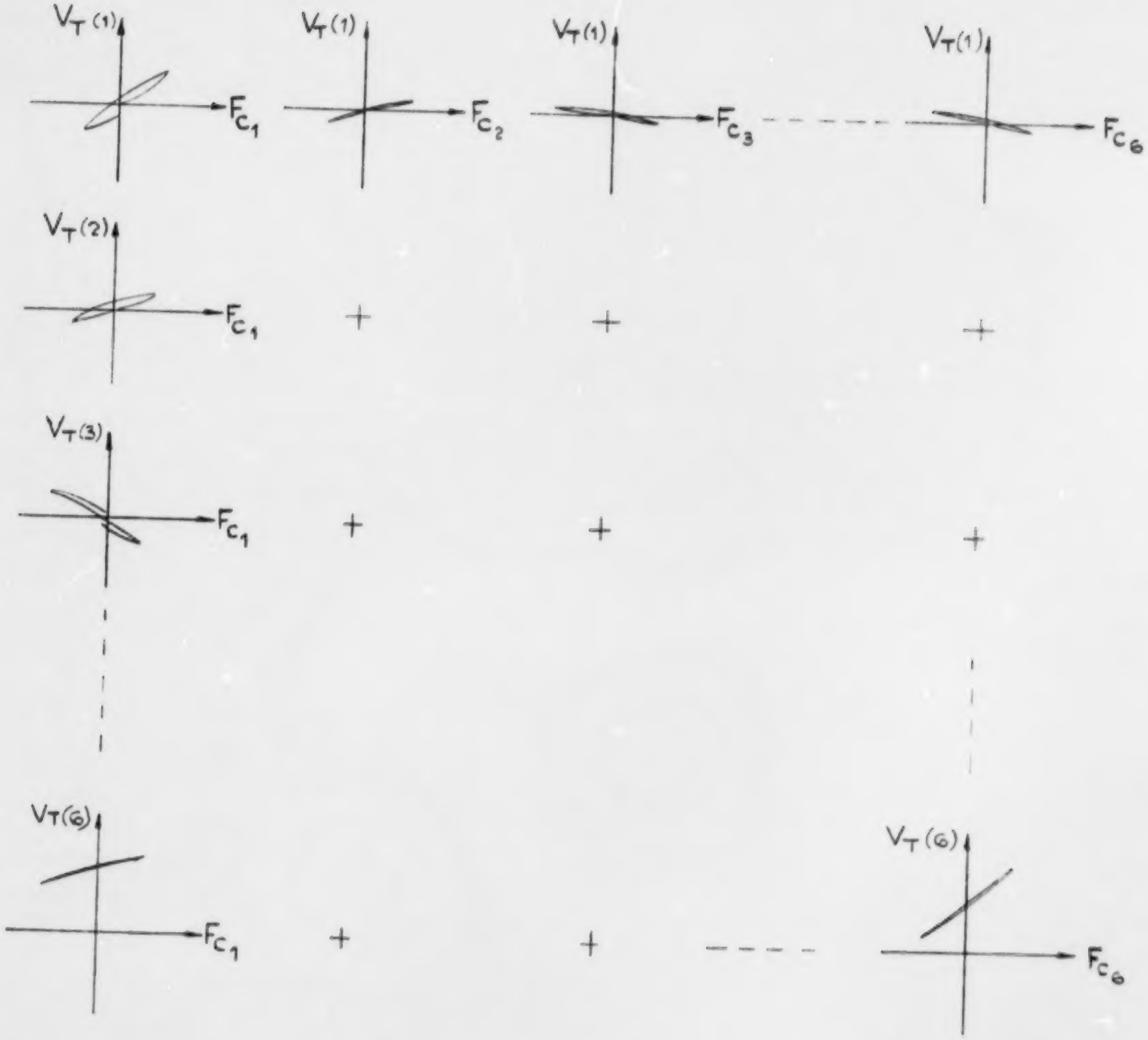

$V_T(1)$
F_{c_1}
$V_T(1)$
F_{c_2}
$V_T(1)$
F_{c_3}
$V_T(1)$
F_{c_6}
$V_T(2)$
F_{c_1}
$V_T(3)$
F_{c_1}
$V_T(6)$
F_{c_1}
$V_T(6)$
F_{c_6}

Figure 2.2-2

CALIBRATION LOAD (LB) $F_{C_{1,K}}$		MEASURED OUTPUTS (A/Ds - No Units)					
		$V_{T(1)}$	$V_{T(2)}$	$V_{T(3)}$	$V_{T(4)}$	$V_{T(5)}$	$V_{T(6)}$
0	1	b_{111}	b_{211}	b_{311}	b_{411}	b_{511}	b_{611}
250	2	.	.	.	.	.	.
500	3	.	.	.	.	.	.
250	4	.	.	.	.	.	.
0	5	b_{112}	b_{212}	b_{312}	b_{412}	b_{512}	b_{612}
-250	6	.	.	.	.	.	.
-500	7	.	.	.	.	.	.
-250	8	.	.	.	.	.	.
0	9	b_{113}	b_{213}	b_{313}	b_{413}	b_{513}	b_{613}
0	1	b_{114}	b_{214}	b_{314}	b_{414}	b_{514}	b_{614}
250	2	.	.	.	.	.	.
500	3	.	.	.	.	.	.
250	4	.	.	.	.	.	.
0	5	b_{115}	b_{215}	b_{315}	b_{415}	b_{515}	b_{615}
-250	6	.	.	.	.	.	.
-500	7	.	.	.	.	.	.
-250	8	.	.	.	.	.	.
0	9*	b_{116}	b_{216}	b_{316}	b_{416}	b_{516}	b_{616}

* These numbers correspond to the points in Fig. 2.2-1

Table 2.2-1: Summary of End to End Data Record for One Calibration Cycle

The letter b_{ijK} is used to denote the ordinate intercept of the calibration curve:

- i identifies the transducer output
- j identifies the externally applied load station
- K denotes the particular reading for the j^{th} external load at the i^{th} transducer

The average value for the ordinate intercept of ${}^{V}T(i)$ is defined as:

$$b_{i1} = \sum_{K=1}^{n} \frac{b_{i1K}}{n} = \overline{b_{i11} + b_{i12} + b_{i13} + \dots b_{i1n}}$$

When a second external load is cycled, say ${}^{F}C_2$, a family of ordinate intercepts b_{i2k} results.

The average is $b_{i2} = \sum_{K=1}^{n} \frac{b_{i2K}}{n}$

The average ordinate intercept for ${}^{V}T(i)$ from all the external load cycles (${}^{F}C_1$, ${}^{F}C_2$, ... ${}^{F}C_6$) is:

$$b_i = \sum_{j=1}^{m} \frac{b_{ij}}{m} = \overline{b_{i1} + b_{i2} + b_{i3} + \dots b_{i6}}$$

In other words, ${}^{V}T(1)$ has an average intercept value of b_1, ${}^{V}T(2)$ intercepts at b_2, etc., where this average is computed as above. The two step process accommodates cases where applied load cycles may not all be of equal length. Computation of b_i can be most effectively accomplished on a calculator after identifying the ordinate data points.

2.3 Slope Calculation

With the intercept defined, the data points for each calibration cycle are used to determine a best fit (least squares method) straight line passing through this intercept. The calculation method is detailed in the reference. Only the intercept values associated with a particular calibration cycle should be used in determining the best fit line. To use intercepts from other calibration cycles for the same transducer output may appear to increase the precision but result in a reduction in the number of degrees of freedom since they have already been included in calculating b_i.

The slopes and intercepts for the calibration may be shown in matrix form as:

$$\begin{bmatrix} V_{T(1)} \\ V_{T(2)} \\ V_{T(3)} \\ V_{T(4)} \\ V_{T(5)} \\ V_{T(6)} \end{bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & C_{14} & C_{15} & C_{16} \\ C_{21} & C_{22} & C_{23} & C_{24} & C_{25} & C_{26} \\ C_{31} & C_{32} & C_{33} & C_{34} & C_{35} & C_{36} \\ C_{41} & C_{42} & C_{43} & C_{44} & C_{45} & C_{46} \\ C_{51} & C_{52} & C_{53} & C_{54} & C_{55} & C_{56} \\ C_{61} & C_{62} & C_{63} & C_{64} & C_{65} & C_{66} \end{bmatrix} \begin{bmatrix} {}^{F}C_1 \\ {}^{F}C_2 \\ {}^{F}C_3 \\ {}^{F}C_4 \\ {}^{F}C_5 \\ {}^{F}C_6 \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \\ b_3 \\ b_4 \\ b_5 \\ b_6 \end{bmatrix}$$

Equation 2.3-1

where $C_{11} = \frac{\partial V_{T(1)}}{\partial F_{C_1}}$ = slope of calibration curve (linearized)

$$C_{42} = \frac{\partial V_{T(4)}}{\partial F_{C_2}}$$

$$C_{53} = \frac{\partial V_{T(5)}}{\partial F_{C_3}} \quad \text{etc.}$$

In matrix notation, this may be written as:

$$\left[V_{Ti}\right] = \left[C_{ij}\right] \left[F_{cj}\right] + \left[b_i\right] \qquad \text{Eq. 2.3-2}$$

and may be compared to $y = mx + b$, the equation for a straight line.

2.4 Force Calibration Errors

The best fit line is constrained to pass thorough the average offset, b_i, and through the data points by the least squares method of curve fitting. Because not all the data points fall on the line, there exists a precision error associated with the linear assumption.

The precision error at any given test point (there are five test points shown in Figure 2.4-1) must be calculated about the assumed line and the worst (i.e., greatest) precision error must be used. This worst case prediction error is defined as δ_{ij}.

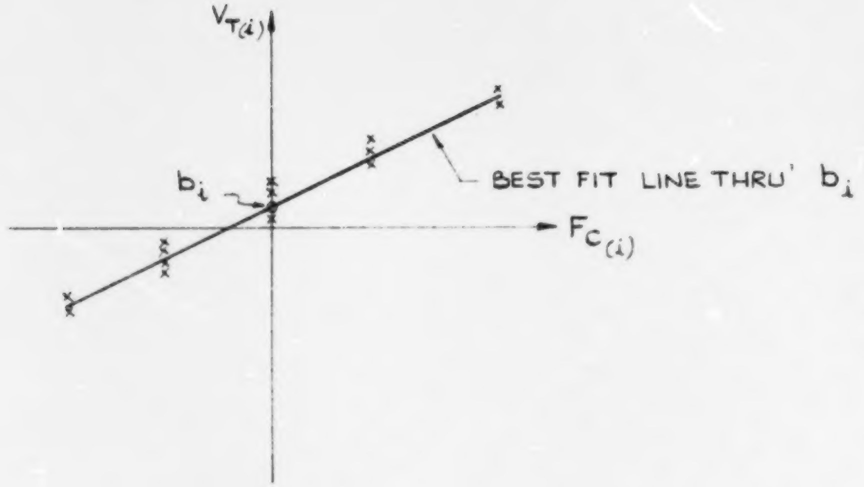


Figure 2.4-1

Expanding the first row of Equation 2.3-1, get:

$$V_{T(1)} = C_{11}F_{C_1} + C_{12}F_{C_2} + C_{13}F_{C_3} + C_{14}F_{C_4} + C_{15}F_{C_5} + C_{16}F_{C_6} + b_1$$

Equation 2.4-1

when the precision errors are added to this expression:

$$V_{T(1)} = C_{11}F_{C_1} \pm \delta_{11} + C_{12}F_{C_2} \pm \delta_{12} + C_{13}F_{C_3} \pm \delta_{13} + C_{14}F_{C_4} \pm \delta_{14} + C_{15}F_{C_5} \pm \delta_{15} + C_6F_{C_6} \pm \delta_{16} + b_1$$

Equation 2.4-2

Following the root sum square method, a single precision error for $V_{T(1)}$ may be calculated as:

$$\Delta_i = \pm \sqrt{\sum_{j=1}^{6} \delta_{ij}^2} \qquad \text{Eq. 2.4-3}$$

or re-writing Eq. 2.3-2 in general terms:

$$\left[V_{Ti}\right] = \left[C_{ij}\right] \left[F_{cj}\right] + \left[b_i\right] \pm \left[\Delta_i\right] \qquad \text{Eq. 2.4-4}$$

Thus the precision error for each transducer output must be calculated. No additional precision error for b_i need be calculated because it is included already in Δ_i.

3.0 ANGLE CALIBRATION OF THE BOGIE

3.1 Pitch Angle (θ) Calibration

Rotation about the lateral axis (y_b) in the positive right-hand sense defines the positive θ or pitch angle. An increase in the height of the front of the trailer with respect to the rear, such as moving from a flat stretch of road to an uphill gradient produces a positive θ. (This should not be confused with a change in relative heights of front and rear on a flat stretch of road, α). See Section 3.4 for α correction.

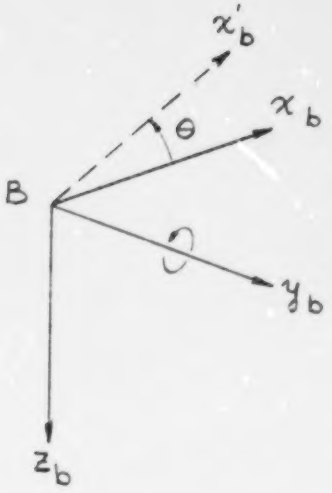
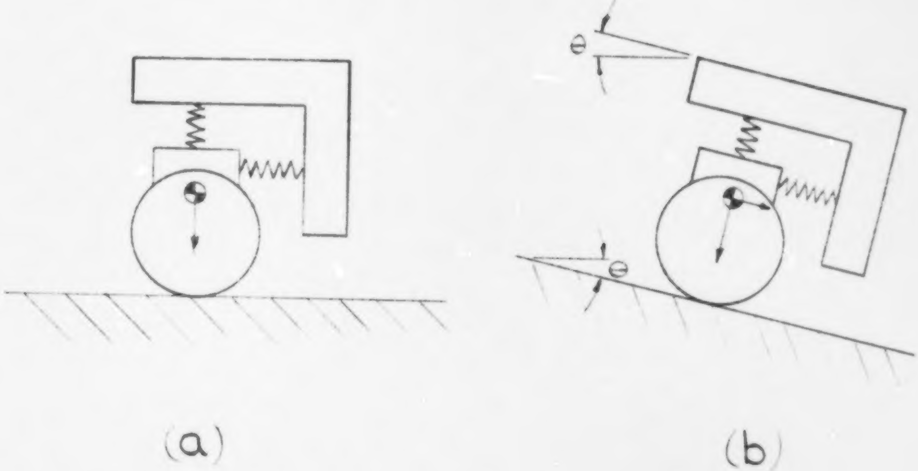

x'_b
θ
x_b
B
y_b
z_b

Figure 3.1-1


θ
θ

(a) (b)

Figure 3.1-2

From Figure 3.1-2, where ⊕ $= W_b + W_w$

$W_b \triangleq$ weight of bogie

$W_w \triangleq$ weight of wheels and bearings, it can be seen that the longitudinal forces are increased by approximately:

$$(W_b + W_w) \sin \theta \doteq (W_b + W_w)\theta$$

while the vertical forces are not reduced, i.e.,

$$(W_b + W_w) \cos \theta \approx (W_b + W_w)$$

In addition, vertical forces at the tire/road interface are increased by the weight of the wheels and bearings, W_w, which is not shown in the calibration procedure. This can be most effectively incorporated in the final vertical load calculations.

The most straightforward way of accommodating variations of θ is through a θ calibration of the bogie on the rig.

The vertical loads are first removed (${}^{F}C_4, {}^{F}C_5$) and at this time the tare weights of the spring/bearing assemblies may be taken.

In the absence of the nominal vertical loads, the offsets of all the load cells will change. The entire rig is now tilted in the θ direction, both positive and negative over the range of the inclinometer, and load cell outputs and the A/Ds are recorded along with inclinometer outputs. Again, good practice dictates that power supply voltages be recorded and raw signal outputs from inclinometers and load cells be recorded in the event any signal processing cards must be replaced.

If at all possible, an independent angle reference should be established; otherwise, any failure of the inclinometer will render the entire calibration process void. Again, nine points per cycle and two cycles are the minimum requirements. Table 3.1-1 shows the data required for calibration and Figure 3.1.3 shows typical output curves.

θ	$V_{T(1)}$	$V_{T(2)}$	$V_{T(3)}$	.....	$V_{T(6)}$
1					
2					
3					
.					
.					
9					
1					
2					
3					
.					
.					
9					

Table 3.1-1

Because the foregoing calibration is performed without wheels and bearings, and to remove the dependency of the calibration on the weight of individual wheelsets and bearings which will

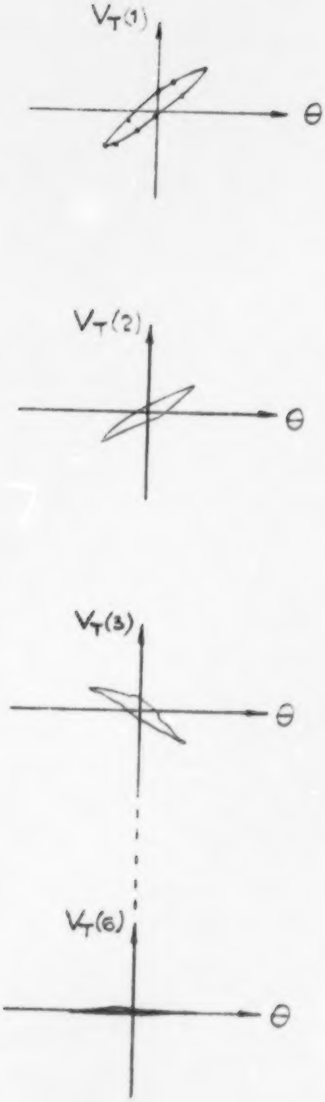

$V_T(1)$
θ
$V_T(2)$
θ
$V_T(3)$
θ
$V_T(6)$
θ

Figure 3.1-3

vary with the tests being performed, a second calibration is performed with wheels and bearings of known weight.

The wheels and bearings to be used are first weighed with a calibrated load cell, then installed on the bogie and the bogie recalibrated through the same nine point two cycle routine as before.

This means a second table of values similar to Table 3.1-1 and a second series of slopes (Figure 3.1-3) will result.

Data reduction consists of calculating the slope of the line for each transducer as a function of θ, (i.e., $\left.\frac{\partial V_{T(i)}}{\partial \theta}\right| W_b$) under the influence of the bogie weight, W_b, alone, repeating the calculation for a bogie and wheel weight, $W_b + W_{w_o}$, and by means of a linear equation, predicting the slope as a function of wheel and bearing weight.

This relation is shown in Figure 3.1-4 and the equation is:

$$C_{\theta i} = M_{o_i} + M_{w_{\theta i}} W_w \qquad \text{Eq. 3.1-1}$$

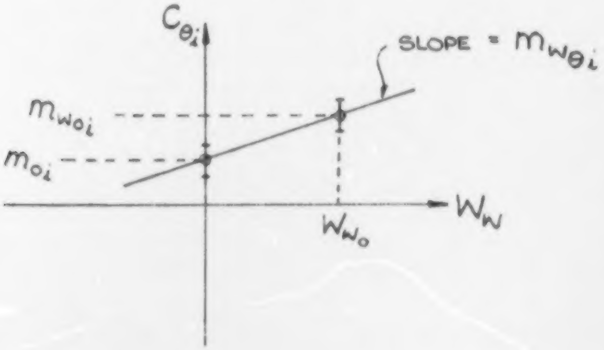


Figure 3.1-4

With reference to Figure 3.1-4, M_{o_i} is the best fit (least squares) slope of the change of $V_{T(i)}$ with θ, $M_{w_{oi}}$ is the best fit (least squares) slope of the change of $V_{T(i)}$ with θ with a wheel and bearing set of weight W_{wo} installed. $M_{w_{\theta i}}$ is the slope (linearized) of the change of $C_{\theta i}$ with W_w between the two points $W_w = 0$ and $W_w = W_{w_o}$.

The precision error associated with the calculation of M_{o_i} is the greatest precision error resulting from the calculation of M_{o_i}, defined as δM_{oi}. The precision error associated with the calculation of $M_{w_{oi}}$ is the greatest precision error resulting from the calibration of $M_{w_{oi}}$, defined as $\delta M_{w_{oi}}$.

The precision error associated with the calculation of $C_{\theta i}$ is the greater of δM_{oi} or $\delta M_{w_{oi}}$, defined as $\delta_{\theta i}$.

Therefore, $b_{\theta i} = C_{\theta i} \; \theta_i \pm \delta_{\theta i}$ Eq. 3.1-2

expresses the offset in every transducer output brought about by a non-zero angle θ. No fixed offset is required because it is already included in b_i.

Equation 3.1-2 is a column vector and may be expanded as:

$$\begin{bmatrix} b_{\theta 1} \\ b_{\theta 2} \\ b_{\theta 3} \\ \vdots \\ b_{\theta 6} \end{bmatrix} = \begin{bmatrix} C_{\theta 1} \\ C_{\theta 2} \\ C_{\theta 3} \\ \vdots \\ C_{\theta 6} \end{bmatrix} [\theta] \pm \begin{bmatrix} \delta_{\theta 1} \\ \delta_{\theta 2} \\ \delta_{\theta 3} \\ \vdots \\ \delta_{\theta 6} \end{bmatrix} \qquad \text{Eq. 3.1-3}$$

When combined with Eq. 2.4-4, it becomes:

$$[V_{Ti}] = [C_{ij}]\ [F_{cj}] + [b_i] + [C_{\theta i}]\theta \pm [\Delta_i] \pm [\delta_{\theta i}]$$

Equation 3.1-4

Again, actual running experience may narrow the range of θ and result in an improvement in accuracy.

3.2 Roll Angle (ϕ) Calibration

Rotation about the longitudinal (x_b) axis in the positive right-hand sense defines the positive ϕ or roll angle. The normal shape of the road from the centreline toward the shoulder gives the bogie a positive roll increment.

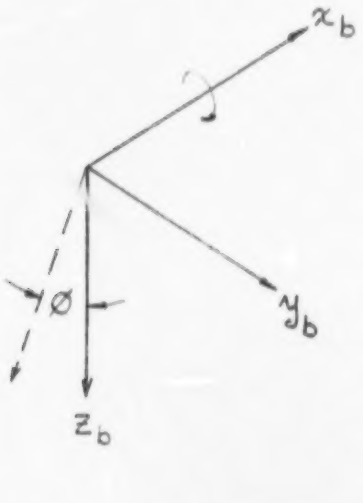


Figure 3.2-1

From Figure 3.1-2 where ⊕ = weight of bogie plus weight of wheels and bearings ($W_b + W_w$), it can be seen that the force at the road in the lateral direction is reduced by approximately

$$(W_b + W_w) \sin \phi \approx (W_b + W_w) \phi$$

while the vertical forces are not reduced

$$\text{i.e., } (W_b + W_w) \cos \phi \approx (W_b + W_w).$$

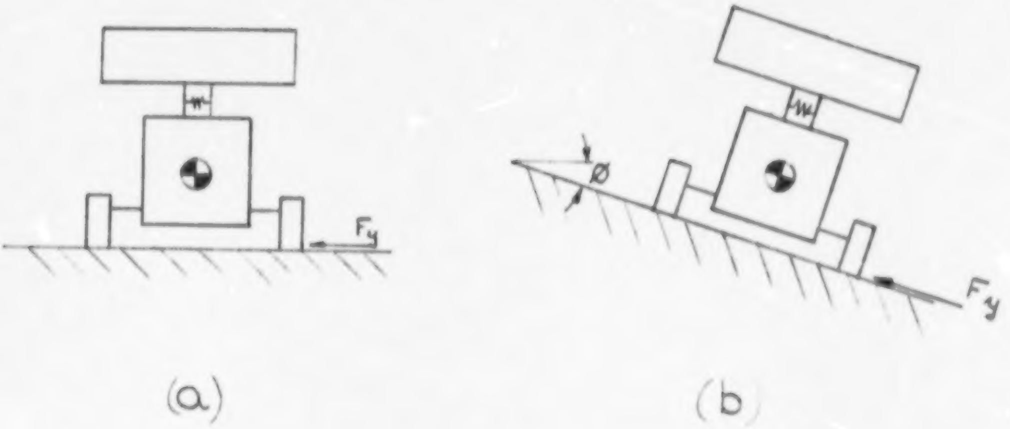


Figure 3.2-2

Roll angle calibration techniques are the same as for pitch angle (Section 3.1). The result is two sets of curves similar to those depicted in Figure 3.1-3. Data reduction consists of calculating the slope of the line for each transducer as a function of ϕ, (i.e., $\left.\frac{\partial V_{T(i)}}{\partial \phi} \right| W_b$) under the influence of the bogie weight, W_b, alone, repeating the calculation for a bogie and wheel weight, $W_b + W_{w_o}$, and by means of a linear equation, predicting the slope as a function of wheel and bearing weight. This relationship is shown in Figure 3.2-3 and the equation is:

$$C_{\phi i} = M_{w_{\phi i}} W_w + M_{o_i} \qquad \text{Eq. 3.2-1}$$

It should be noted that a positive roll angle tends to decrease the apparent road force in the lateral direction. For this reason, a sign inversion is required for the lateral coefficient of equation 3.2-1.

That is, $C_{\phi 6} = -[M_{w_{\phi i}} W_w + M_{o_i}]$

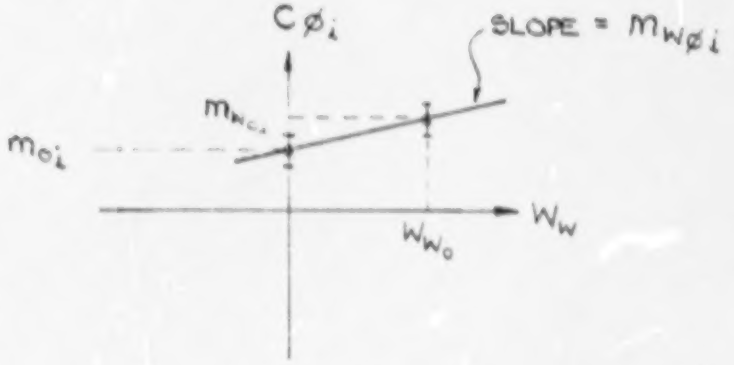


Figure 3.2-3

With reference to Figure 3.2-3, M_{o_i} is the best fit (least squares) slope of the change of $V_{T(i)}$ with ϕ, $M_{w_{\phi i}}$ is the best fit (least squares) slope of the change of $V_{T(i)}$ with ϕ with a wheel and bearing set of weight W_{w_o} installed.

$M_{w_{\phi i}}$ is the slope (linearized) of the change of $C_{\phi i}$ with W_w between the points $W_w = 0$ and $W_w = W_{w_o}$. The precision error associated with the calculation of M_{o_i} is the greatest precision error resulting from the calculation of M_{o_i}, defined as ${}^{\delta}M_{oi}$. The precision error associated with the calculation of $M_{w_{oi}}$, defined as ${}^{\delta}M_{w_{oi}}$. The precision error associated with the calculation of $C_{\phi i}$ is the greater of ${}^{\delta}M_{oi}$ or ${}^{\delta}M_{w_{oi}}$, defined as $\delta_{\phi i}$.

The offset in every transducer output brought about by a non-zero angle ϕ is:

$$b_{\phi i} = C_{\phi i}\phi_i \pm \delta_{\phi i} \qquad \text{Eq. 3.2-2}$$

Equation 3.2-2 is a column vector and may be expanded as:

$$\begin{bmatrix} b_{\phi 1} \\ b_{\phi 2} \\ \cdot \\ \cdot \\ \cdot \\ b_{\phi 6} \end{bmatrix} = \begin{bmatrix} C_{\phi 1} \\ C_{\phi 2} \\ \cdot \\ \cdot \\ \cdot \\ C_{\phi 6} \end{bmatrix} \left[\phi\right] \pm \begin{bmatrix} \delta_{\phi 1} \\ \delta_{\phi 2} \\ \cdot \\ \cdot \\ \cdot \\ \delta_{\phi 6} \end{bmatrix} \qquad \text{Eq. 3.2-3}$$

When combined with equation 3.1-4 it becomes:

$$\left[V_{Ti}\right] = \left[C_{ij}\right]\left[F_{cj}\right] + \left[b_i\right] + \left[C_{\theta i}\right]\theta + \left[C_{\phi i}\right]\phi \pm \left[\Delta_i\right] \pm \left[\delta_{\theta i}\right] \pm \left[\delta_{\phi i}\right]$$

Equation 3.2-4

3.3 Trailer Inclination (α) Calibration

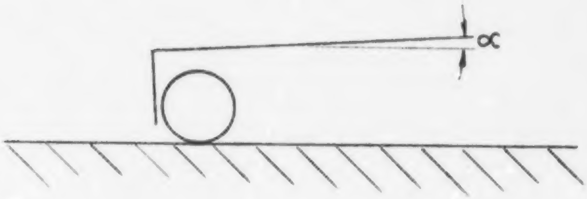


Figure 3.3-1

A positive trailer inclination angle α results when the front of the trailer rises with respect to the rear, relative to the nominal "0" condition established by the longitudinal inclinometer measurement θ during bogie calibration. Vertical loads on the trailer are now transmitted to the longitudinal load cells as the product of the sine of the inclination of the trailer and the vertical load. Because vertical loads are large with respect to the longitudinal forces being measured, this cross coupling term must be treated very accurately if meaningful data is to be obtained. During the initial setup prior to road test, the trailer should be adjusted for 0 α as far as possible by shimming the bogie or axle, or by reducing tire pressure in the tractor. While running, tire growth with heat buildup or changes in trailer load may cause variations in α.

Trailer inclination is measured by two trailing wheels. Calibration of these wheels will be dealt with later on and in this section it will be assumed an α measurement exists.

From the derivation of the equations for the force bridge, it can be seen that a positive α increments the longitudinal forces by an amount,

$$\Delta F_{x_o} = \alpha(F_{Z_R} + F_{Z_L})$$

where $\sin\alpha \approx \alpha$ is assumed. This correction is a product of variables and cannot be linearized as with the previous terms.

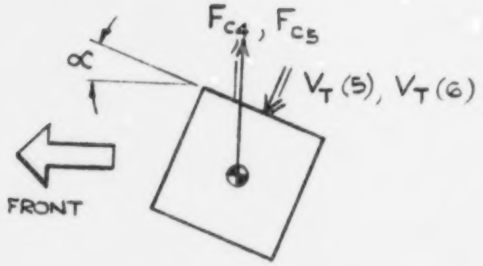


Figure 3.3-2

Calibration is as follows:

1) Set F_{C_4}, F_{C_5} to nominal vertical loads and set α to zero as measured by the θ inclinometer. The transducer outputs should be equal to the column vector $[b_i]$ and this set of data points may be used to refine the average b_i values.

2) Tilt the calibration rig to an angle α, and realign the vertical calibration forces ${}^{F}C_4, {}^{F}C_5$ to be plumb. Adjust these forces so that b_5 and b_6 return to their nominal values (i.e., $V_{T(5)}$, $V_{T(6)}$).

3) For $V_{T(1)}$ through $V_{T(4)}$, record outputs. Again transducer outputs, A/D outputs, and power supply voltages should be recorded.

4) Repeat this procedure over the range of α for one cycle. A second cycle should be repeated after steps 5, 6, and 7.

5) Increase the nominal load on ${}^{F}C_4, {}^{F}C_5$ at $\alpha = 0$ and record $V_{T(1)}$ through $V_{T(6)}$.

6) Cycle α through same values as before, maintaining $V_{T(5)}$ and $V_{T(6)}$ at the values determined in step 5).

7) Repeat steps 5) and 6) at other values of ${}^{F}C_4$, ${}^{F}C_5$.

8) Repeat entire process to obtain additional cycles.

The actual number of calibration measurements made will depend on the observed repeatability, linearity, and to some extent, time available.

If the data so obtained were plotted, a map as shown in Figure 3.3-3 would result from each of $V_{T(1)}$ through $V_{T(4)}$.

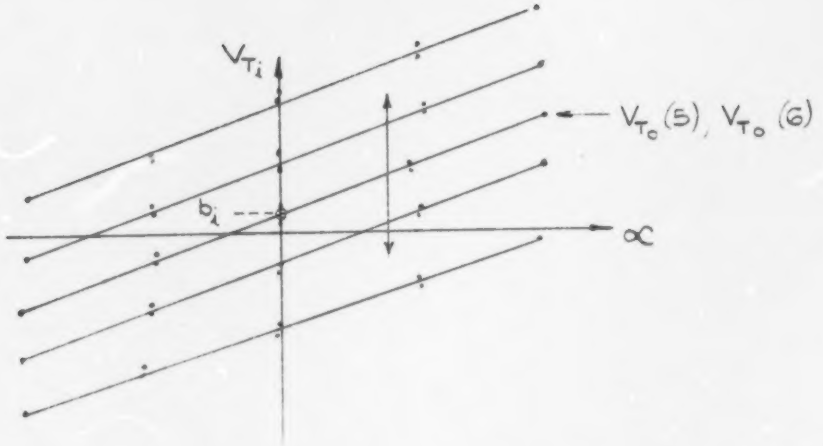


Figure 3.3-3

Each of the lines on this map represents a linearization of the data from each cycle to the same slope with the constraint that the nominal load line pass through b_i. Call this slope $C_{\alpha i}$.

The equation representing this map is given as:

$$V_{T(i)} = C_{\alpha i}\alpha + b_i + K_i\{(V_{T(5)} + V_{T(6)}) - (V_{To(5)} + V_{To(6)})\}$$

Equation 3.3-1

where K_i is the best fit slope through the ordinate intercept data of the parameter. Because the slopes are constant a group of ordinate intercepts may be used.

The precision error $\delta_{\alpha i}$ is the greatest precision error of the data groups anywhere on the map about the best fit equation 3.3-1.

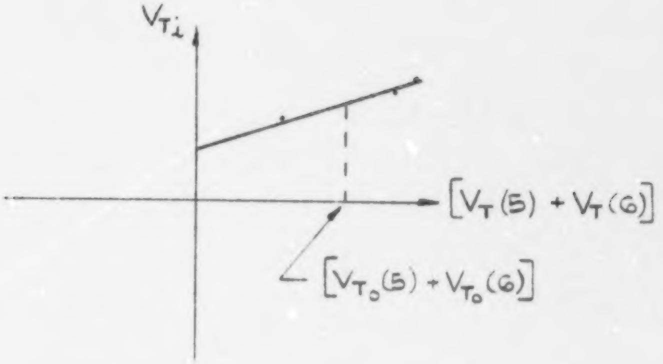


Figure 3.3-4

The incremental adjustment to the transducer readings is expressed as:

$$b_i = C_{\alpha i}\alpha + K_i(V_{T(5)} + V_{T(6)}) - b_i^* \pm \delta_{\alpha i} \qquad \text{Eq. 3.3-2}$$

where $b_i^* = K_i(V_{To(5)} + V_{To(6)})$ Eq. 3.3-3

Incorporating Eq. 3.3-2 into Eq. 3.2-4:

$$\begin{aligned}[V_{Ti}] = {} & [C_{ij}][F_{cj}] + [b_i] + [C_{\theta i}]\theta + [C_{\phi i}]\phi + [C_{\alpha i}]\alpha \\ & + [K_i]\,(V_{T(5)} + V_{T(6)}) - [b_i^*] \\ & \pm [\Delta_i] \pm [\delta_{\theta i}] \pm [\delta_{\phi i}] \pm [\delta_{\alpha i}] \qquad \text{Eq. 3.3-4}\end{aligned}$$

Treatment of the error associated with measuring α is contained in Section 3.5.

3.4 Pitch Angle Correction

Because the longitudinal inclinometer uses a gravity reference to establish its reading, it does not truly represent road gradients in the presence of a trailer inclination. Where θ_M is the measured or gravitational inclination of the bogie, the true θ may be calculated as:

$$\theta = (\theta_M - \alpha) \qquad \text{Eq. 3.4-1}$$

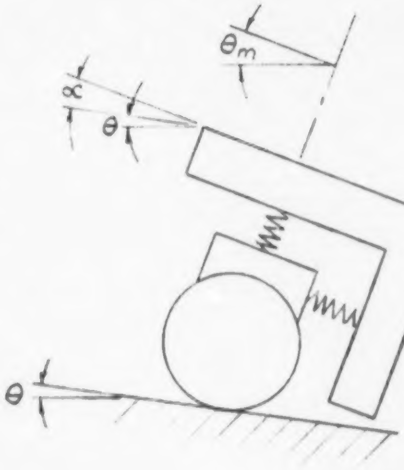
Measurement of θ_M and ϕ is to greater accuracy than can be expected with the instrumentation and so no further errors are considered for these angles. This is not the case with the α measurement. Defining Δ_α as the precision error associated with the measurement of α (Section 3.5 provides the derivation) and substituting Eq. 3.4-1 into 3.3-4:

$$\begin{aligned}\left[V_{Ti}\right] = {} & \left[C_{ij}\right]\left[F_{cj}\right] + \left[b_i\right] + \left[C_{\theta i}\right](\theta_M - \alpha \pm \Delta_\alpha) + \left[C_{\phi i}\right]\phi \\ & + \left[C_{\alpha i}\right](\alpha \pm \Delta_\alpha) + \left[K_i\right](V_{T(5)} + V_{T(6)}) - \left[b_i{}^*\right] \\ & \pm \left[\Delta_i\right] \pm \left[\delta_{\theta i}\right] \pm \left[\delta_{\phi i}\right] \pm \left[\delta_{\alpha i}\right] \qquad \text{Eq. 3.4-1}\end{aligned}$$

This can be regrouped as follows:

$$\begin{aligned}\left[V_{Ti}\right] = {} & \left[C_{ij}\right]\left[F_{cj}\right] + \left[b_i\right] + \left[C_{\theta i}\right]\theta_M + \left[C_{\phi i}\right]\phi + \left[C_{\alpha i} - C_{\theta i}\right]\alpha \\ & + \left[K_i\right](V_{T(5)} + V_{T(6)}) - \left[b_i{}^*\right] \\ & \pm \left[\Delta_i\right] \pm \left[\delta_{\theta i}\right] \pm \left[\delta_{\phi i}\right] \pm \left[\delta_{\alpha i}\right] \pm \left[C_{\theta i}\right]\Delta_\alpha \pm \left[C_{\alpha i}\right]\Delta_\alpha\end{aligned}$$

Equation 3.4-2



θ_m
α
θ
θ

Figure 3.4-1

3.5 TARR Calibration

Calibration of the Tangent Angle-Rolling Radius measurement system is straightforward. It is performed as the last step before road tests with the bogie, tires, and TARR installed in the trailer. The θ inclinometer is used as the calibration reference and no error is assigned to it due to its high accuracy. The tires and TARR wheels must be on a flat surface and a floor jack may be used to raise and lower the front of the trailer. Measurement of the angle α is made with two LVDTs attached to the TARR wheel assemblies as shown in Figure 3.5-1.

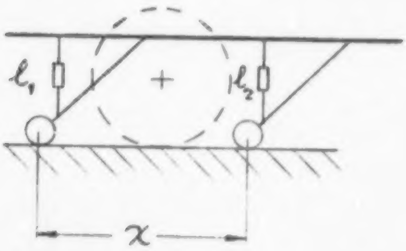


Figure 3.5-1

α is computed as $\frac{\ell_1 - \ell_2}{x}$. The electronics Eq. 3.5-1 will subtract the digital outputs corresponding to the LVDT readings and by measuring θ_M over a range of values and recording the difference between ℓ_1 and ℓ_2 readings, the calibration is established. The procedure should be repeated after raising both measuring wheels an equal increment, and again after raising the tires a fixed increment.

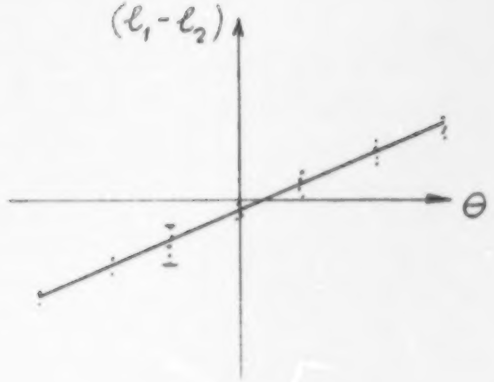


Figure 3.5-2

All $(\ell_1 - \ell_2)$ values as a function of θ_M may be used in the calculation of a least squares linear approximation and again the worst case precision error is selected as δ_T.

The equations are:

$$(\ell_1 - \ell_2) = M_T \theta_M + b_T \pm \delta_T \qquad \text{Eq. 3.5-2}$$

$$\text{Therefore } \alpha = \{(\ell_1 - \ell_2) - b_T \pm \delta_T\} \frac{1}{M_T} \qquad \text{Eq. 3.5-3}$$

$$\text{or } \alpha = \frac{(\ell_1 - \ell_2) - b_T}{M_T} \pm \frac{\delta_T}{M_T} \qquad \text{Eq. 3.5-4}$$

$$\text{Therefore } \Delta_\alpha = \pm \frac{\delta_T}{M_T} \qquad \text{Eq. 3.5-5}$$

Because of the sensitivity of all results to the measured value of α, a high degree of precision should be sought in the foregoing calibration.

Effective rolling radius, r_e, is calculated as:

$$r_e = \frac{\ell_1 + \ell_2}{2} + r_{e_o} \qquad \text{Eq. 3.5-6}$$

Calibration is straightforward and consists of displacing both measuring wheels equal amounts and recording the $(\ell_1 + \ell_2)$ term and accurately measuring and recording distance of axle ℄ to flat surface. This calibration cycle should be repeated over the range of α as measured by θ_M, and all data used in reduction and error assignment. Jacks or tire inflation, or blocks, may be used to vary ℓ_1, ℓ_2.

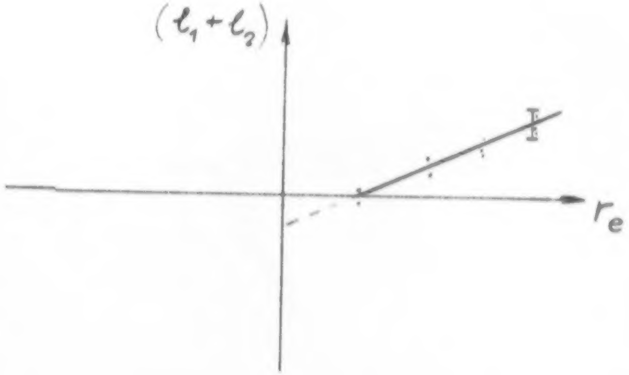


Figure 3.5-3

The equations follow.

$$(\ell_1 + \ell_2) = M_r \, r_e + b_r \pm \delta_r \qquad \text{Eq. 3.5-7}$$

Therefore $$r_e = \frac{(\ell_1 + \ell_2) - b_r}{M_r} \pm \frac{\delta_r}{M_r} \qquad \text{Eq. 3.5-8}$$

$$\Delta_r = \frac{\delta_r}{M_r} \qquad \text{Eq. 3.5-9}$$

4.0 CALIBRATION MATRIX

4.1 Error Reduction

The error components associated with the coefficient matrix are:

$$\pm\left[\Delta_i\right] \pm \left[\delta_{\theta i}\right] \pm \left[\delta_{\phi i}\right] \pm \left[\delta_{\alpha i}\right] \pm \left[C_{\theta i}\right]\Delta_\alpha \pm \left[C_{\alpha i}\right]\Delta_\alpha$$

These are the precision errors from the calibration data measurement and reduction. Bias errors are not included in the error analysis because of the difficulty in putting a reliable estimate on their magnitude. Bias errors will result in erroneous absolute rolling resistance measurements but relative differences will be significant to the degree permitted by the precision error.

Statistical theory postulates that the precision error will be normally distributed in terms of probability. The chances of all error sources being at their maximum at the same time is extremely remote.

The method most commonly used to calculate an effective error is called the ROOT SUM SQUARE (RSS) method. A single precision error is calculated by taking the square root of the squares of the individual error terms. Defining $\S$, as the combined error term,

$$[\S_i] = \mathrm{RSS}([\Delta_i],[\delta_{\theta i}],[\delta_{\phi i}],[\delta_{\alpha i}],[C_{\theta i}]\Delta_\alpha,[C_{\alpha i}]\Delta_\alpha) \qquad \text{Eq. 4.1-1}$$

Equation 4.1-1 is expanded for clarity:

$$\S_1 = \{\Delta_1{}^2 + \delta_{\theta 1}{}^2 + \delta_{\phi 1}{}^2 + \delta_{\alpha 1}{}^2 + (C_{\theta 1}\Delta_\alpha)^2 + (C_{\alpha 1}\Delta_\alpha)^2\}^{\frac{1}{2}}$$

$$\S_2 = \{\Delta_2{}^2 + \delta_{\theta 2}{}^2 + \delta_{\phi 2}{}^2 + \delta_{\alpha 2}{}^2 + (C_{\theta 2}\Delta_\alpha)^2 + (C_{\alpha 2}\Delta_\alpha)^2\}^{\frac{1}{2}}$$

$$\vdots$$

$$\S_6 = \{\Delta_6{}^2 + \delta_{\theta 6}{}^2 + \delta_{\phi 6}{}^2 + \delta_{\alpha 6}{}^2 + (C_{\theta 6}\Delta_\alpha)^2 + (C_{\alpha 6}\Delta_\alpha)^2\}^{\frac{1}{2}}$$

Equation 4.1-2

Thus, a single column vector may be used to replace all the error terms of Eq. 3.4-2 giving:

$$\begin{aligned}[V_{Ti}] = {} & [C_{ij}][F_{cj}] + [b_i] + [C_{\theta i}]\theta_M + [C_{\phi i}]\phi + [C_{\alpha i} - C_{\theta i}]\alpha \\ & + [K_i](V_{T(5)} + V_{T(6)}) - [b_i{}^*] \pm [\S_i] \qquad \text{Eq. 4.1-3}\end{aligned}$$

4.2 Generating the Calibration Matrix

The influence coefficient matrix derived in the preceding sections has established the dependency of transducer readings on forces applied in a road oriented reference frame. This influence coefficient matrix must be inverted so that forces in the road oriented reference frame may be determined from transducer outputs. The road reference frame forces are termed F_{o_i}. The form of Eq. 4.1-3 may be described as:

Transducer Outputs Equal:

- the influence coefficient matrix times the calibration forces;
- plus transducer output modifiers as a function of transducer outputs ($\theta_M, \phi, \alpha, V_{T(5)}, V_{T(6)}$); and,
- a precision error term.

Inversion of this expression begins with grouping the transducer terms and errors on one side:

$$[V_{Ti}] - [b_i] - [C_{\theta i}]\theta_M - [C_{\phi i}]\phi - [C_{\alpha i} - C_{\theta i}]\alpha - [K_i](V_{T(5)} + V_{T(6)})$$
$$+ [b_i{}^*] \pm [\S_i] = [C_{ij}][F_{cj}] \qquad \text{Eq. 4.2-1}$$

Define the inverse of the coefficient matrix as D:

$$\text{i.e., } D = C^{-1}$$
$$\text{or, } [d_{ji}] = [C_{ij}]^{-1} \qquad \text{Eq. 4.2-2}$$

TP 3652

SNOWMOBILES AS UTILITY VEHICLES IN THE ARCTIC: CURRENT PERFORMANCE AND PROSPECTS FOR IMPROVEMENT.

CA1
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Prepared for:
The Transportation Development Centre
Transport Canada

May 1982

MM

TP 3652

Transport Canada / Transports Canada
Library and Information Centre /
Centre de Documentation GSLA
Ottawa, Canada K1A 0N5

SNOWMOBILES AS UTILITY VEHICLES IN THE ARCTIC: CURRENT PERFORMANCE AND PROSPECTS FOR IMPROVEMENT.

Final Report

Dr. P.J. Poole

May 1982

The contents of this report reflect the views of the contractor and are not necessarily the official views or policies of the Transportation Development Centre.

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REPORT DOCUMENTATION FORM

1. Transport Canada Report No. TP3652	2. TDC Project No. 4281	3. Recipient's Catalogue No. -
4. Title and Subtitle SNOWMOBILES AS UTILITY VEHICLES IN THE ARCTIC: Current Performance and Prospects for Improvement.		5. Report Date May 1982
		6. Performing Organization Report No. -
7. Author(s) P.J. Poole Ph.D.		8. Transport Canada File No. D1465-320
9. Performing Organization Name and Address Peter Poole Consulting Services R.R.1. Dunham Quebec, J0E 1M0.		10. DSS File No. T8200-0-0535
		11. DSS or Transport Canada Contract No. OSD81-00037
12. Sponsoring Agency Name and Address Transportation Development Centre (TDC) 1000 Sherbrooke St. W. P.O. Box 549, Place de l'aviation Montreal, Quebec, H3A 2R3		13. Type of Report and Period Covered Final: April 81 -May 82
		14. Sponsoring Agency Code 4281
15. Supplementary Notes Summary reports: In French under TP3728F In English under TP3728E		16. TDC Project Officer J.E. Laframboise

17. Abstract

A research study precipitated by reports of escalating costs of snowmobile acquisition and operation in the Canadian Arctic, accompanied by decreasing utility as working machines. The objective was to estimate the prospects for developing a more suitable machine for Inuit use. Historical surveys revealed, declining sales, fewer manufacturers and models, and a convergence in design since 1970 - with present sales about 25% of that peak year. The Inuit market is only 1% of the total but is comparatively stable. Technical research indicated that accumulated engineering knowledge is sufficient to design a suitable utility machine but such a model would be unlikely to appeal to the mass of recreational users. Moreover, current commitments to large volume production prohibit small-volume specialised manufacture at reasonable cost. As an example, a general description was given of a utility machine with the emphasis upon pulling power and durability and a simplicity of construction based upon a modular approach to design and assembly. It was concluded that the economic/design problems posed by present methods of mass production might be circumvented by evolving a production/assembly process viable for local, small-scale manufacture. An investigation into the feasibility of such a course of action was recommended as a follow-up to this study. It was further suggested that this be undertaken as a collaboration between a team of engineers and representative Inuit organisations.

18. Key Words Snowmobiles Arctic Transportation		19. Distribution Statement Limited number of copies available at TDC	
20. Security Classification (of this report) nil	21. Security Classification (of this page) nil	22. No. of Pages 144	23. Price -

TDC/CDT 79-005

FORMULE DE DOCUMENTATION POUR RAPPORT

1. Nº du rapport Transports Canada: TP3652

2. Nº de l'étude — CDT: 4281

3. Nº de catalogue du destinataire: -

4. Titre et sous-titre:
SNOWMOBILES AS UTILITY VEHICLES IN THE ARCTIC:
Current Performance and Prospects for
Improvement (Full Report)

5. Date du rapport: mai 1982

6. Nº du rapport de l'organisme exécutant: -

7. Auteur(s): P.J. Poole Ph.D.

8. Nº de dossier Transports Canada: D1465-320

9. Nom et adresse de l'organisme exécutant:
Peter Poole Consulting Services
R.R.1. Dunham
Quebec, J0E 1M0

10. Nº de dossier — ASC: T8200-0-535

11. Nº de contrat — ASC ou Transports Canada: SD81-00037

12. Nom et adresse de l'organisme parrain:
Centre de développement des transports (CDT)
1000 O., rue Sherbrooke
C.P. 549, Place de l'aviation
Montréal, Québec, H3A 2R3

13. Genre de rapport et période visée: Final: avril 81-mai 82

14. Code de l'organisme parrain: 4281

15. Remarques additionnelles:
Rapport sommaire: en français sous TP3728F
en anglais sous TP3728E

16. Agent de projet — CDT: J.E. Laframboise

17. Résumé

Cette étude fait suite à l'observation du coût croissant et de l'utilité décroissante des motoneiges utilisées dans l'Arctique. On a voulu évaluer les possibilités de production d'un véhicule mieux adapté aux besoins des Inuits. Une revue du marché de la motoneige indique une décroissance des ventes ainsi qu'une réduction des modèles disponibles et des manufacturiers: les ventes sont présentement à 25% du niveau de 1970. Le marché Inuit n'en représente que 1% mais demeure stable. L'étude démontre que l'expertise existe pour la conception d'une motoneige arctique, mais que ce véhicule utilitaire ne pourrait plaire à la majorité des utilisateurs sportifs. De plus, les impératifs industriels ne permettraient pas la production à un prix raisonnable d'un véhicule à diffusion restreinte. Pour illustration le rapport définit les spécifications générales d'une motoneige utilitaire fondées sur une construction modulaire, en mettant l'accent sur la capacité de traction, la durabilité et la simplicité.

Afin de contourner le dilemme quantité/coût il est proposé d'étudier un procédé d'assemblage faisant appel à la main-d'oeuvre Inuit. Ceci ferait l'objet d'une étude de faisabilité en collaboration avec des associations Inuits.

18. Mots-clés:
Motoneige

Transport arctique

19. Diffusion:
Nombre limité de copies
disponible au CDT

20. Classification de sécurité (de ce rapport): nil

21. Classification de sécurité (de cette page): nil

22. Nombre de pages: 144

23. Prix: -

CDT TDC 79-005

ACKNOWLEDGEMENTS

The author gratefully acknowledges the guidance and assistance of staff of the Inuit Development Corporation and the Kativik Regional Government and, in addition, the special contributions of Willie Emudluk and Jean-Guy St. Aubin, George River, Quebec.

Also, the following members of the project Steering Committee contributed by periodically reviewing progress:

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FOREWORD

The issue of escalating costs of snowmobile acquisition and operation in the Canadian Arctic, without concommittant improvements in utility performance, was bought to light in the course of investigations by the Inuit Development Corporation into the prospects for expanding intersettlement trade between Inuit communities. The snowmobile/sled combination was one means of transport considered for this purpose.

Officials from the Arctic Transportation Directorate and the Transportation Development Centre expressed an interest in the work of the Inuit Development Corporation. The Transportation Development Centre responding to the "snowmobile problem" by sponsoring a project to investigate ways of resolving this issue. The Inuit Development Corporation was represented on the inter-departmental committee which reviewed progress on the project and. in addition, provided active assistance and encouragement while it was in progress.

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SUMMARY OF CONCLUSIONS AND RECOMMENDATIONS

Conclusions

1. The design and production of snowmobiles is certainly dominated by the need to serve the recreational market, but a long-term, and continuing decline in that market is obliging manufacturers to pay more attention to the requirements of the very small but comparatively stable Arctic utility market.

2. This market is now being served by modifying certain recreational models and presenting them as "dual-purpose" machines. Thus, the potential for responding to utility requirements is limited by the scope these recreational models provide for modification.

3. The modifications undertaken by manufacturers are limited in that sense and also by the economics of production: how to produce small volume models or features at acceptable prices.

4. The methods by which snowmobiles are now produced prohibit the possibility of designing and manufacturing an entirely new utility snowmobile. For such purposes, users must rely upon two utility models that have been in production for over ten years.

5. The developments and changes that have occurred in the course of refining snowmobiles for recreational use have not always been disadvantageous for utility use, but, it appears that certain avenues of research which could have improved utility performance may have been inevitably passed by in favour of those with more promise for recreational performance.

6. The dual-purpose models recently introduced are providing improved utility performance, but not at lower cost. In fact, the continuing escalation in costs of aquisition and operation is likely to become the dominant factor in selecting future snowmobiles.

7. There may be scope for the small-scale production of a mid-size utility snowmobile if this were done by a simplified modular process of assembly. Such a machine might follow early snowmobiles in basic concept but would take advantage of improved technology and materials. It would not possess the speed or styling typical of current models.

Recommendations

1. If Conclusion No. 7 is to be acted on, the first step should be a study of its practical and economic feasibility and a comparison of the results with the production economics of present manufacturers.

2. This first step in a possible R&D program should only be undertaken if an Inuit or other organisation with a committed interest in the further development of the high Arctic assumes responsibility for coordinating such a program and acting at the interface between technologists and Inuit users.

1. INTRODUCTION

> Over the past years the different manufacturers have invested untold moneys developing stable, fast, comfortable snowmobiles. Weight has always been a factor in gaining the final results. The lighter the final production unit is, the faster it eats up the trails, manoeuver turns and moguls and result in an overall more comfortable riding machine.
>
> The original production machines that most of us cut our teeth on could be described as nothing more than *anchors*. They were assembled with heavy materials for a good reason - the type of terrain the sleds were called upon to cover. As much as we hate to admit it, before the trails systems became as expansive as they are today, *the open fields, lakes and yes, even paved roads* were the main snowmobile trails. . . .
>
> In order to stand up to this type of punishment, the sleds were *over-built*. *Today, what might be a thin aluminium bracket, had to be made of steel plate 15 years ago*.
>
> "Snowmobile Canada", 1982, 13(2):42-43.
> (emphasis added)

The sentiments expressed in this quotation, and the snowmobile design changes they reflect, have precipitated the study described in this report. Fifteen to twenty years ago, the "anchors" were the mainstay of the snowmobile market. But this market was limited to those who required snowmobiles as working or utility machines. But the demand for utility machines was soon overtaken by an escalating demand for snowmobiles as recreational machines and, once the basic design configuration became established, further technical evolution was dominated by the requirements for accomplishing well-organised travel along well-groomed trails with style, speed, comfort and safety. In the course of this design evolution, the requirements for snowmobiles as utility machines were largely neglected and, until very recently, the only advertised working machines on the market were two survivals from the era of the "anchors".

The unsuitability of recreational snowmobiles when put to working uses has been demonstrated most emphatically in the tundra regions north of the tree-line. In these extreme operating environments, snowmobiles are subjected to more demanding uses by Inuit, in the course of hunting, trapping and fishing, than by any other group of users. In the late 1970's the Board of Directors of the Fédération des Coopératives du Nouveau Québec formally requested one manufacturer to re-introduce a model which had acquired a good reputation as a working machine but which had been withdrawn from the production. The inadequacies of currently available models were repeatedly brought to the attention of staff of the Inuit Development Corporation engaged in programs of renewable resource development. They were also noted at a, 1978, meeting of the Science and Technology Committee of the Advisory Committee on Northern Development, when the Chairman of the Northwest Territories Science Advisory Board, Dr. O. Solandt, observed that "A better and more rugged skidoo is needed that is not a recreational type and sells for half the current price" (Eggleton 1978). This remark summarises the two principle dimensions of the 'snowmobile problem': (1) the question of machine design and construction, (2) the problem of rising costs for initial purchase and spare parts - exacerbated by escalating costs for fuel. This observation also begs the question of whether both problems can be resolved simultaneously; this question is the central economic issue addressed by this study.

1.1. STUDY OBJECTIVES

The Overall Objective contained in the Work Statement for this project is the "Development of a vehicle for light transportation tasks in the Arctic having an improved performance compared with present snowmobiles."

"The project is being undertaken by the Transportation Development Centre (TDC) of Transport Canada in collaboration with the Arctic Transportation Directorate (DGRT) to obtain a detailed understanding of the problem and evaluate the feasibility of an R&D program to improve snowmobile performance" (TDC 1981).

At first glance, it would appear that the overall objective stated above might be attained in one of three ways:

1. By determining snowmobile improvements that may readily be made to existing models without radical changes in production procedures. These would be design changes which would not increase production costs to the point where market research would be required to support their adoption.
2. By determining major improvements, which would require significant changes in tooling and production economics and in turn require confidence on the part of the manufacturer that the market would bear the eventual increased price to the buyer.
3. By determining that the kind of improvements required cannot be realistically expected to be developed by

snowmobile manufacturers, given the small size of the Arctic utility market, and could therefore only be provided by a government supported R&D program. This, of course, would not necessarily resolve the question of how the more costly improvements emerging from such a program could be economically incorporated in snowmobile production.

All three statements imply that the problem is one of production engineering as much as design engineering R&D and, in fact, as this study progressed, the question of the economic feasibility of snowmobile improvements emerged as a dominant theme. The manufacturers know how to make more rugged machines; they do not feel that the small utility market will accept the costs.

The suggestion that snowmobiles should be built that are both more rugged and less expensive reflects the view that, by simply discarding those 'recreational features', that have added to costs while actually degrading utility performance, this objective may be brought closer. This may be an understandable reaction to some of the more flashy and costly amongst currently available models; however, it should be remembered that the utility models that have survived from the sixties are not significantly different in price from other models. The case may rather be that snowmobiles have become more expensive for similar reasons that cars and motorcycles have, and that, just as aircraft built and equipped for arctic operation are more expensive than their southern counterparts, so will utility snowmobiles inevitably cost more.

1.2. STUDY AREA AND LIMITATIONS

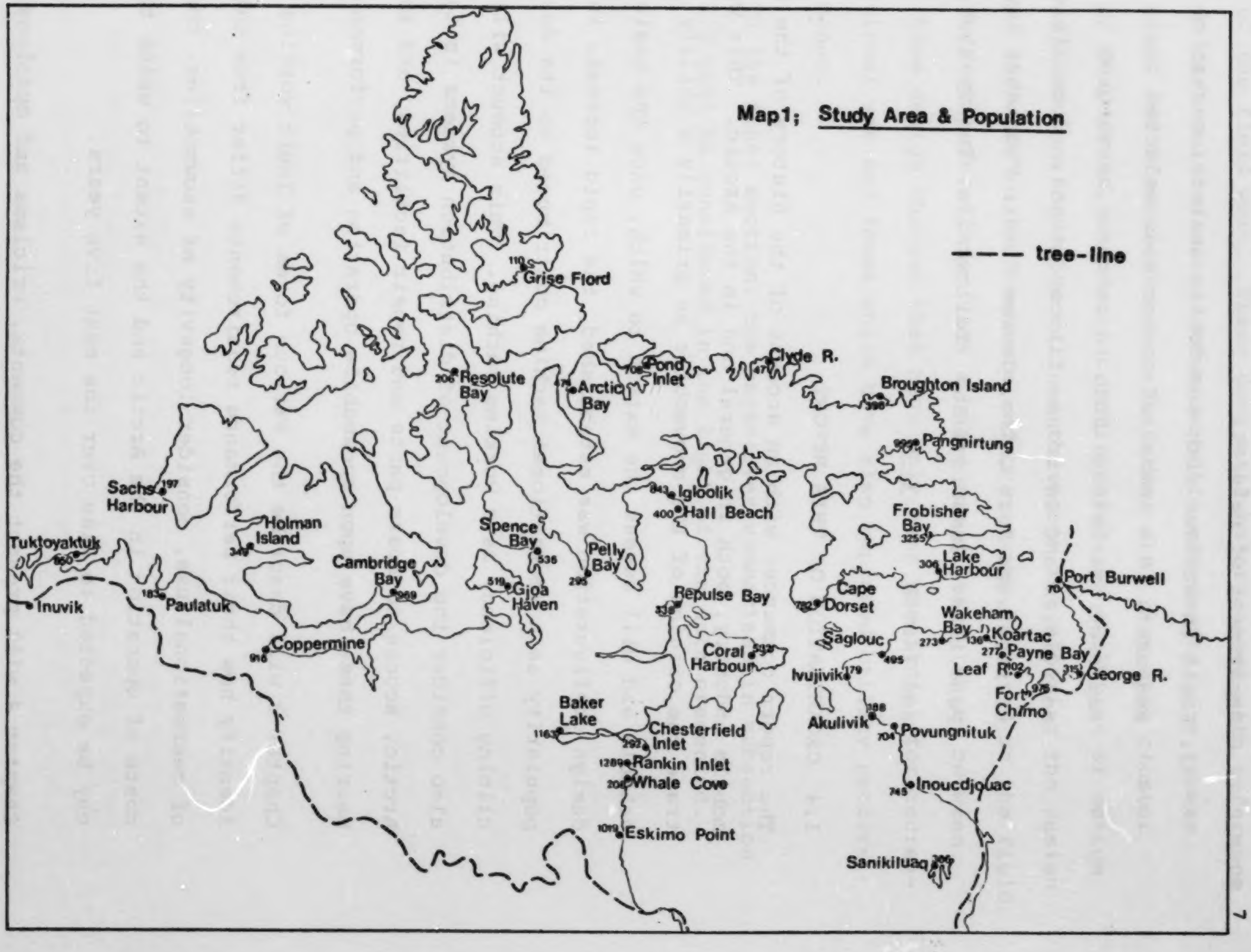
The study will concentrate upon snowmobile use in Canada north of the tree-line in the Northwest Territories and Québec (Map 1). Thus, the principle focus will be upon light vehicle transport over tundra snow and the landfast ice. However, since many of the inhabitants of communities at or close to the tree-line may regularly travel into the taiga region, this use of snowmobiles amongst trees and in comparatively deep and soft snow will also be taken into consideration in this context.

The study will concentrate further upon the use of snowmobiles by Inuit in this region for hunting, trapping, fishing and long-distance travel. With relatively minor exceptions, such as reindeer herding in Tuktoyaktuk, these four activities encompass the range of uses to which snowmobiles are put. Other user groups, for example scientific parties or the armed forces, may also rely upon snowmobiles as working vehicles, but the indications are that such uses do not depart significantly from Inuit use, either in terms of intensity of use or precise type of activity. In fact, the material gathered in this study suggests that what may be called serious or regular use of snowmobiles by Inuit places more demands upon the machinery than any other type of utility use.

1.3. STUDY METHODOLOGY

The study was conducted in three phases:

1. Library research on previous studies on snowmobile performance and records of snowmobile performance maintained by government agencies - including the



Map1; Study Area & Population

---- tree-line
110 Grise Fiord
206 Resolute Bay
475 Arctic Bay
708 Pond Inlet
477 Clyde R.
390 Broughton Island
999 Pangnirtung
843 Igloolik
400 Hall Beach
197 Sachs Harbour
349 Holman Island
660 Tuktoyaktuk
Inuvik
183 Paulatuk
Cambridge Bay 969
Spence Bay 536
519 Gjoa Haven
295 Pelly Bay
916 Coppermine
339 Repulse Bay
Coral Harbour 532
782 Cape Dorset
3255 Frobisher Bay
306 Lake Harbour
70 Port Burwell
273 Wakeham Bay
136 Koartac
277 Payne Bay
Leaf R. 102
151 George R.
975 Fort Chimo
Saglouc 495
Ivujivik 179
188 Akulivik
704 Povungnituk
745 Inoucdjouac
Sanikiluaq 296
1163 Baker Lake
292 Chesterfield Inlet
1289 Rankin Inlet
204 Whale Cove
1019 Eskimo Point

Department of Defense.

2. Field interviews with snowmobile dealers, mechanics and users in a number of communities selected to represent variations both in resource harvesting activities and environmental conditions. Communities visited were Fort Chimo, George River, Frobisher Bay, Pond Inlet, Cambridge Bay Yellowknife, Tuktoyaktuk, and Inuvik.

1.4 ORGANISATION OF THIS REPORT

The report commences with an account of the history of the snowmobile industry, both in general and in the Arctic. This will trace the origins of the snowmobile as primarily a utility vehicle and will assess the extent to which, once the basic design configuration was established, the rapid increase in its popularity as a recreational machine contributed to its declining efficiency as a working machine. This account will also consider the development of distribution systems in the Arctic, access to spare parts and repair facilities, and the bearing these have upon snowmobile operation and performance.

Chapter 3 will describe the various forms of Inuit working use, identify how their performance requirements differ from those of recreational use, consider longevity of snowmobiles, the costs of operation in the Arctic and the extent to which these may be expected to rise over the next five years.

Chapter 4 will present the comments, cricisms and opinions by users, mechanics and dealers that were gathered in the course of the field work. These data will be organised with reference to particular clusters of snowmobile components, e.g. power plant and transmission, tracks/skis, etc. For each cluster of components, the factors influencing the evolution of design will be considered as well as the scope for altering the design in response to the comments and criticisms gathered in the field. This chapter will provide a basis for distinguishing between those design changes that have only been beneficial for recreational use and those which have also improved utility performance.

In the final section, the objectives summarised in subsection 1.3 will be considered in the light of the data presented.

2. THE SNOWMOBILE INDUSTRY

> Polar Snowmobile, 1961, price: $1,495, specifications: rear mounted 9.5 hp engine, welded 16 and 18 guage steel body, windshield, 2 inch padded seat, brake, electrical system (starter-generator, lights, battery), clutch guard, first-aid kit, snowshoe rack, compass. Weight: 660 lbs. Estimated price in 1981: United States dollars, $3-4,000.
>
> Snowmobile Industry & Sport
> Newsletter
> March/April 1981, p. 19.

2.1. INTRODUCTION

This chapter commences with an historical account of the changes that have taken place in the snowmobile industry since its beginnings in the 1960's. This account illustrates how dramatic sales increases in the recreational market led manufacturers to evolve designs which responded to the demand from that sector. Such design development was not inevitably antithetical to the needs for Arctic utility machines, but any benefits this market derived from that evolution were largely incidental.

By examining the past relationship between the Arctic utility market and the southern recreational market, this chapter will attempt to determine the extent to which the Arctic utility sector is dependent upon the recreational sector. This will provide a background for evaluating the influence that current trends will exert upon the future development of utility machines.

The first of the following sections will review progress since the 1960's and will review the general merits of current snowmobiles models as utility machines. The second will review the organisation of the snowmobile industry north of the tree-line and discuss the efforts that have been made by manufacturers to produce a machine suitable for use in that region.

2.2. THE SNOWMOBILE INDUSTRY: 1960's TO THE PRESENT

Four significant trends appear during this period: (1) a dramatic increase and then slower decline in snowmobile sales; (2) a gradual decrease in the number of snowmobile manufacturers and number of machines produced; (3) a trend towards uniformity in the design of snowmobiles, and (4) an increase in Japanese participation in the industry.

2.2.1. Snowmobile sales since the 1960's: Canada and the United States

Table 1 has been produced from data provided by the International Snowmobile Industry Association (ISIA). It seems that comprehensive sales records were not maintained before 1964 and those between 1964 and 1967 were subjected to some degree of extrapolation. However, they are detailed enough to illustrate the magnitude of the 200% increase in total sales that occurred during the 1968/69 season, an increase which continued, though less dramatically, until 1970/71. High sales levels, approaching 500,000 units a year persisted until 1972/73, after which they suffered a 29% decline, a trend which continued until sales feel below 200,000 units in 1976/77. Sales

Table 1: RECORD OF SNOWMOBILE SALES IN THE UNITED STATES, CANADA, ALASKA, YUKON & NORTHWEST TERRITORIES, 1964-1982

Year	Total Sales	United States	Canada	Alaska	Yukon & NWT	Alaska Yukon & NWT
until 1964	35,000	23,870	11,130	478	56	534
64/65	40,000	27,280	12,720	546	140	686
65/66	55,000	37,510	17,490	751	297	1,048
66/67	70,000	47,740	22,260	956	512	1,468
67/68	85,016	58,095	26,921	1,164	790	1,954
68/69	255,051	174,286	80,765	3,491	2,370	5,861
69/70	423,427	296,604	126,823	4,927	2,999	7,926
70/71	493,662	342,035	151,627	4,829	2,683	7,512
71/72	458,162	332,177	125,985	4,735	1,715	6,450
72/73	449,570	316,036	133,534	4,956	836	5,792
73/74	443,044	290,624	144,420	6,660	1,051	7,711
74/75	316,014	192,185	123,829	4,031	527	4,558
75/76	242,043	153,902	88,741	4,391	435	4,826
76/77	194,887	111,489	83,398	4,034	2,081	6,115
77/78	226,035	140,550	85,484	4,638	2,403	7,041
78/79	260,154	174,644	85,510	4,653	1,718	6,371
79/80	189,826	127,175	82,651	4,103	1,764	5,867
80/81	166,168	99,047	67,121	3,904	1,995	5,899
81/82	(147,000) est.	N/A	N/A	N/A	N/A	N/A

recovered slightly over the next two seasons, though they barely passed the 250,000 level; they then declined to levels below those which preceded the enormous surge in 1968/69. The latest (total only) shipment figures received by ISIA (to January 31, 1982) are 120,000. It appears unlikely that this total will exceed 130,000 for 1981/82, a further decline of 22% over the previous season.

Figure 1 illustrates the relationship between Canadian sales and United States sales over this period. This shows that though sales in both countries accelerated at comparable rates during the boom years of 1967 to 1971, subsequent behaviour varied, with the United States' market both exhibiting wider fluctuations than the Canadian market and eventually falling more steeply to constitute a smaller share of the total market than it had held during the peak sales season of 1971/71. During 1970/71 the United States share was 69%, by 1980/81 it had fallen to 59%. With declining sales, a similar change has taken place in the relationship between total sales and those in the Yukon, the Northwest Territories and Alaska. This change is reviewed below.

2.2.2. Snowmobiles shipments since the 1960's: Northwest Territories, Yukon and Alaska

Figure 2 shows the shipment levels since the 1960's. These are again extracted from ISIA records, which give only combined figures for the NWT and the Yukon. It is interesting to note that, while shipments to northern regions shared the enormous

Figure 1

SNOWMOBILE SHIPMENTS, 1964-1981: NORTH AMERICA, UNITED STATES, CANADA

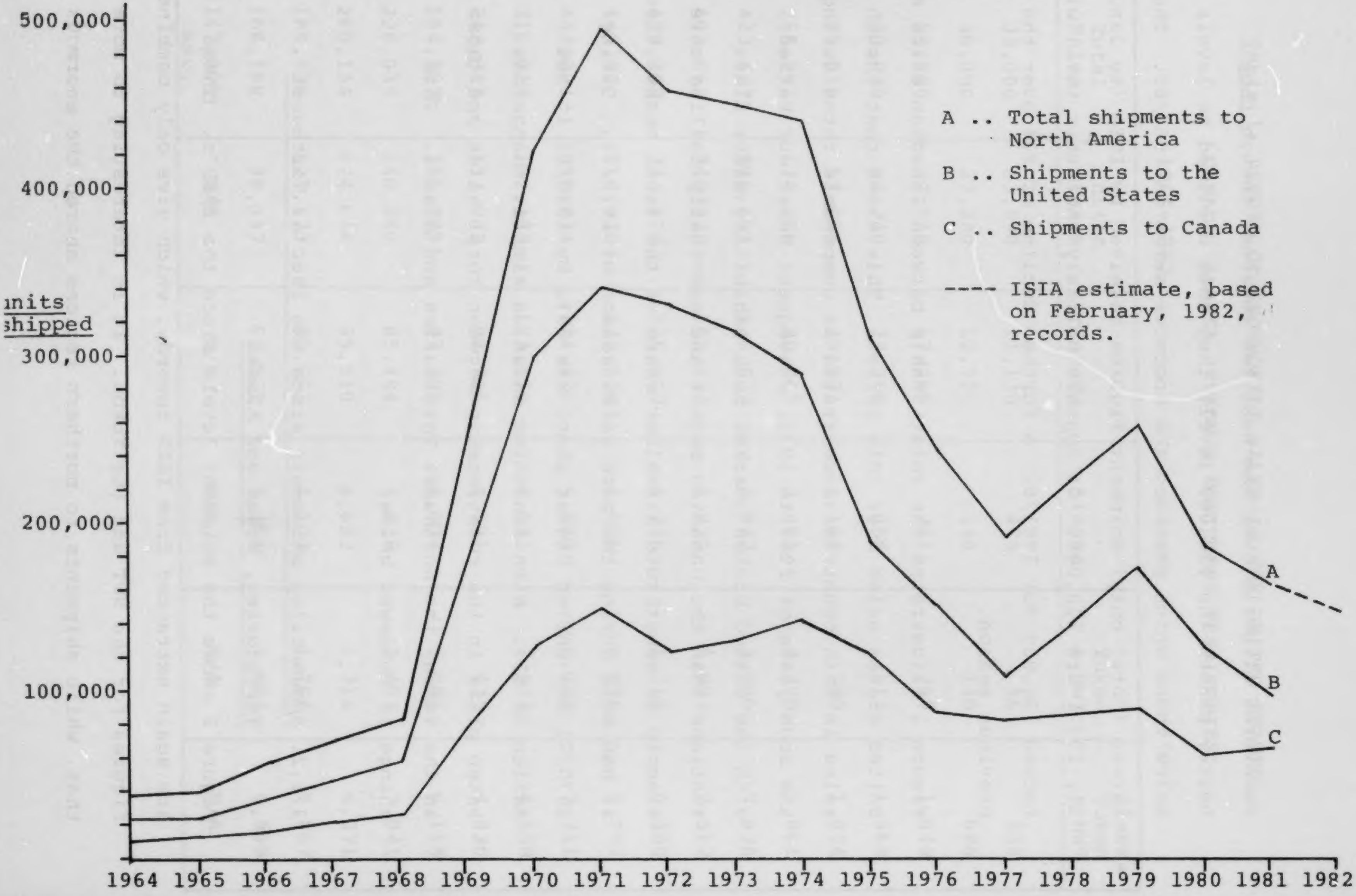
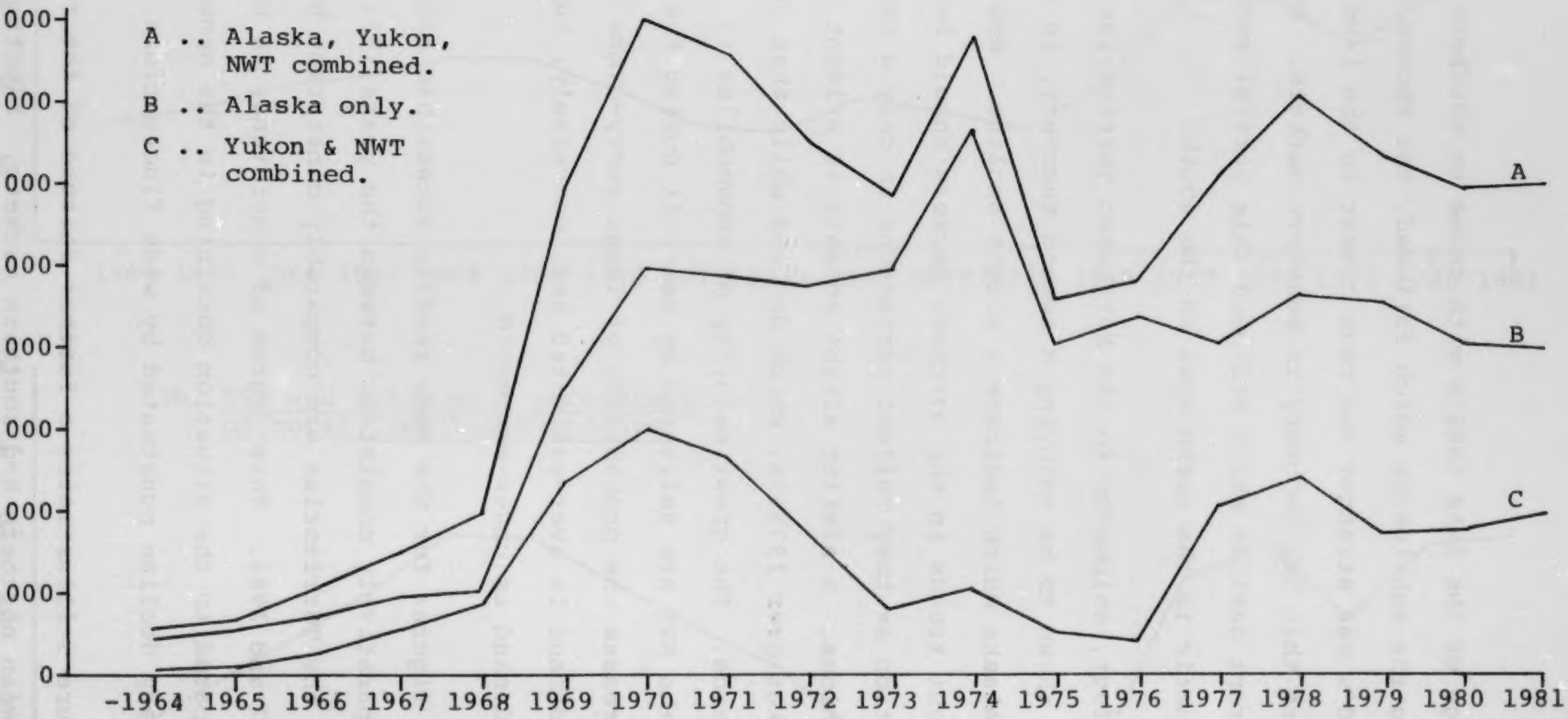


Figure 2

SHIPMENTS OF SNOWMOBILES TO NWT, YUKON, ALASKA, 1964-1981



surge of the late 1960's with those to southern regions, and also the subsidence which followed, the recovery in the late 1970's was stronger and came closer to the levels of the peak years than the recovery in southern markets. Furthermore, the current decline which followed this partial recovery was less dramatic in the north than in the south.

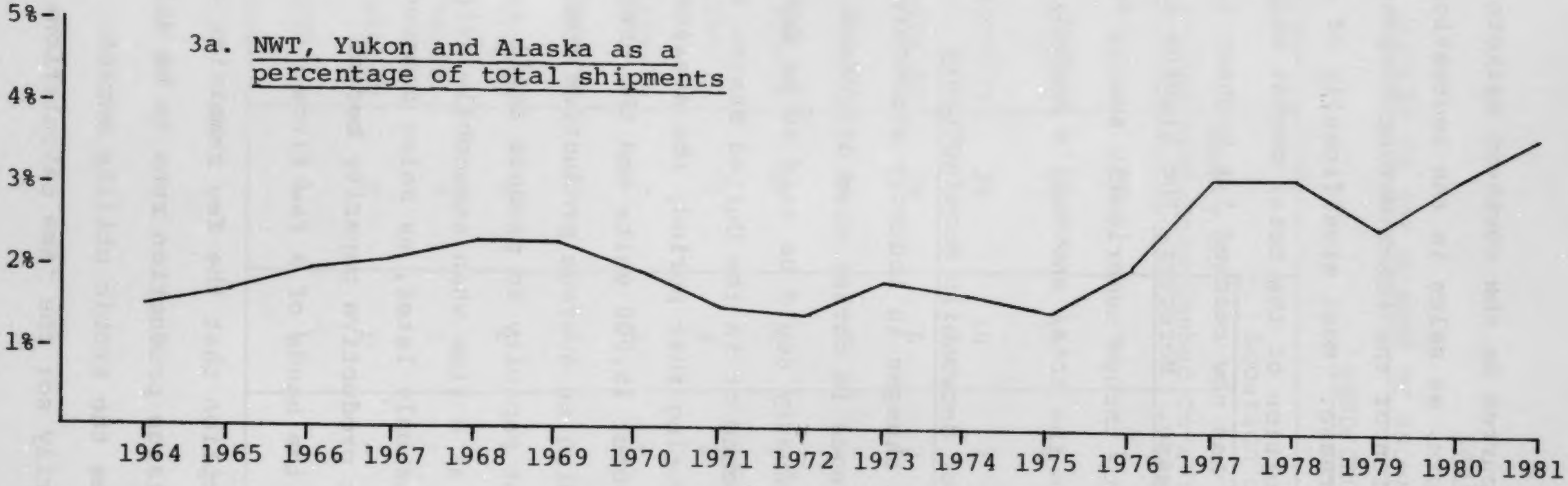
In fact, shipments to the Northwest Territories and the Yukon are shown to be enjoying a second recovery, in contrast to those in Alaska which indicate a slight decline. However, such slight trends in the northern markets should be treated with caution as they reflect variations in only a few hundred machines. A similar slight anomaly is evident in the NWT/Yukon market over 1975-76, which dropped while that in Alaska turned upwards. The great majority of snowmobiles to Inuit communities in the NWT are delivered by sea-lift during the summer. This increases the probability of large carry-overs of inventory, if demand is over-estimated and, conversely, suppressed sales, if demand is under-estimated.

The figures for the more readily accessible Alaska market are comparatively consistent between the years 1975 and 1981, those for the Territories are comparably consistent between the years 1977 and 1981. This degree of consistency is noticeable when compared to the situation obtaining in the general market - a steady decline punctuated by wide fluctuations.

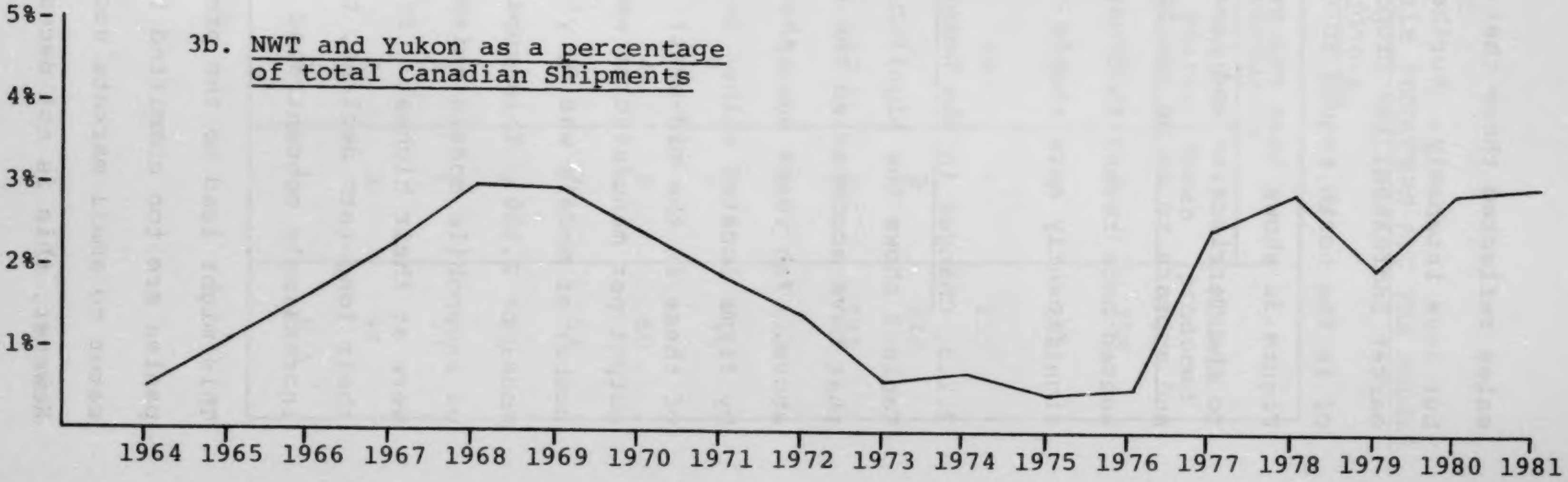
Figure 3 illustrates a further feature of the relationship between northern and southern markets. Fluctuations in northern

Figure 3

SNOWMOBILE SHIPMENTS TO NORTHERN MARKETS AS PERCENTAGES OF TOTAL SNOWMOBILE SHIPMENTS



3a. NWT, Yukon and Alaska as a percentage of total shipments



3b. NWT and Yukon as a percentage of total Canadian Shipments sales reflected those that occurred in the southern markets, but less intensely. Furthermore, as sales in the recreational market increased, the proportion of the total market disposed of in the north tended to decrease. Most significantly of all, Figure 3a shows that the proportion of the total market shipped to the Territories and Alaska has now reached its highest level and appears to be on the increase. Moreover, the figures presented here reveal this market, though numerically small, to be significantly more stable than the total snowmobile market.

2.2.3. Changes in the Number of Snowmobile Manufacturers

Table 2 shows the significant changes in industry structure that have accompanied the changes in market size discussed above. Ten years ago, the industry could be said to be dominated by firms located either in Quebec or in the United States (most of these in the mid-west). During that period, the average output per manufacturer was about 15,000 units and the average number of models was 6, yielding an average production run per model of 2,500. This apparent capacity to produce small runs of snowmobile models existed at a time when snowmobile sales were at their highest. It was only later, as sales commenced their long-term decline, that productive capacity became increasingly concentrated in the hands of a few firms.

This might lead to the presumption that the few remaining companies are too committed to large production runs to be able to cater to small markets such as the Arctic utility market. However, this is not necessarily so; the same calculations

Table 2

CHANGES IN THE NUMBER OF SNOWMOBILE PRODUCERS AND THE MODELS PRODUCED, 1970-82

Season	Number of producers	Location of producers				Models produced
		Quebec	Ontario	United States	Japan	
1970/71	38	8	1	28	1	258
1971/72	27	12	2	12	1	248
1972/73	28	10	1	16	1	235
1973/74	29	8	2	17	2	222
1974/75	23	8	1	13	1	145
1975/76	18	6	1	10	1	121
1976/77	10	1	1	6	2	75
1977/78	7	1	-	4	2	60
1978/79	7	1	-	4	2	54
1979/80	7	1	-	4	2	61
1980/81	7	1	-	4	2	54
1981/82	5	1	-	2	2	44
1982/83	4	1	-	2	1	N/A

applied to last year's figures for models and units produced yield an average of 4,000 units per model. What is at stake may be not so much a manufacturer's need to have large runs to justify production of any single model as the need for large total sales - of all models produced. Also of significance is whether models are distinguished 'cosmetically' or by basic structural character; this question will be discussed further in Chapter 4.

2.2.4. Explanations given for recent decline in total sales

A number of explanations have been proffered at one time or another either to explain the general sales decline since the peak of 1970/71 or the more recent decline which followed the apparent recovery between the years 1977 and 1979. Whether or not the present trend will continue is of importance for the utility snowmobile market for the greater the general decline, the more likely that manufacturers will pay close attention to the demands of Arctic utility users. The factors which are believed to be responsible for the decline in sales are summarized below.

Economic factors often cited are: the increasing costs of fuel, the increasing costs of snowmobiles themselves, a general withdrawal from large 'luxury' purchasing, high interest rates, recession. It should be mentioned that estimates of the annual costs of fuel for recreational snowmobiling are in the area of $200, a relatively small amount in comparison with the capital and depreciating costs of snowmobiling. Some dealers whose opinions were canvassed in this study expressed the view that those people who already possess snowmobiles have found it cheaper to overhaul present machines rather than trade them in for new models. IASA reports that there is virtually no second hand trade in snowmobiles through dealers; most of such sales are made privately.

A second cluster of explanations refers to the status of snowmobiling as a sport. During the early 1970's when sales were at their highest, the industry endured a series of regulatory battles, on the questions of noise emissions, safety, and environmental impact. Though the industry survived, by reducing noise levels, enforcing safety requirements and going to considerable lengths to confine snowmobile travel to recognised trail systems, it is arguable that the sport only survived with a certain stigma attached - and significantly reduced scope for attracting new adherents. While snowmobiling was struggling to establish itself as a legitimate sport, cross-country skiing has emerged as a major participant sport, partly in reaction to escalating costs of downhill skiing, partly as part of broadening interest in physical fitness sports.

Not only has snowmobiling been affected by growing interests in cross-country skiing, but, in the area of mechanised sports, is now facing direct competition from the use of three-wheeled all terrain vehicles. These vehicles (see Annex 1) are capable of traversing groomed snowmobile trails and have done so to such an extent that ISIA has recommended that these machines be excluded from the trails as the two vehicles are not compatible on such prepared surfaces. Sales of off-road three-wheelers were estimated at 80,000 in 1979 and 100,000 in 1980 (ISIA 1981). These machines tend to be more economical, cost less, and, most of all, can be used all year over a wider variety of terrain types than snowmobiles. Respondents to the survey undertaken by ISIA acknowledged that, apart from their choice of machine, the riders of three-wheelers and snowmobiles had much in common. Undoubtedly, such machines offer a powerful alternative attraction to those persons inclined towards mechanised recreation who can afford it.

Dealers and manufacturers consulted in this study partially attributed the sales decline since 1979 to two poor snow winters in succession. This winter, 1981/82, has, since late December become an excellent snow winter and the manufacturers are hoping that sales, which were conforming to the recent pattern of decline until December, will revive in the new year and justify the view that availability of snow is the overriding factor determining sales levels.

At the time of completion of this study, total shipment figures have become available to the end of February 1982. These are 141,000. They strongly indicate a continuation of the long-term decline in sales, despite this winter's generally excellent snow conditions.

2.2.5. The significance for the utility market of the decline in sales

It would be tempting to conclude that the decline in total sales will oblige manufacturers to take more seriously the requirements of the utility market, as it assumes a progressively larger proportion of the total market. On the other hand, the decline may continue to the point where the production of any kind of snowmobile becomes economically unfeasible at current price levels.

In fact the first point appears already to have been reached, and manufacturers have begun to take account of the small but stable Arctic market. When questioned about what they intended to do by way of serving it, manufacturers made quite positive declarations, either of an intention to produce suitable utility models (see subsection 2.3.4. below), or of an intention not to do so on economic grounds.

Those manufacturers who have decided to cultivate the Arctic market have generally done so by modifying current production machines rather than by designing models that are radically different. The former is regarded as the more cost-effective approach, but it, of course, raises the question of the extent to which the 'ideal' utility machine can be put together from elements primarily designed for recreational snowmobiling. Chapter 4 will deal with this problem in more detail and will attempt to distinguish between utility improvements which are within the scope of existing production tools and methods and those which would require expensive re-tooling.

2.2.6. Currently available snowmobile models

To complete this historical account of the snowmobile industry in general, a listing is provided in Table 3 of those models currently available. The table includes Arctic Cat and Scorpion 1980/81 models as these are still available this season and because the future of these model ranges continues to remain in doubt.

Selected models have been annotated and will be re-considered in greater detail at the close of chapter 4 (see Table 9). Those models that are annotated fall into four general groups:

*A - Models which for a long period have been produced as utility machines.

*B - Recent models, advertised as utility or dual purpose machines.

*C - Models which, though not designed primarily as utility snowmobiles, have acquired a reputation as dependable working machines.

*D - The smallest and cheapest models.

These models will be combined later in Table 9 with other projected or prototype utility models in an attempt to arrive at a profile of the most practical working snowmobile.

When the two *A type utility machines, the Bombardier Elan and Alpine, are excluded from consideration, the data listed in Table 2 indicate the extent to which there has been a convergence in snowmobile design since the basic configuration became established in the late 1960's. All the remaining models employ slider suspension (with the exception of the Scorpions, now out of production); the great majority use capacitor discharge

Table 3: CURRENTLY AVAILABLE SNOWMOBILE MODELS AND THEIR MAJOR ATTRIBUTES

MODEL NAME	MAJOR DESIGN FEATURES											
	Weight k	Displacement cc	No. Cylinders	No. Carburetors	Cooling (A) air (F) fan (L) liquid	Starting (M) manual (E) electric E/M both	Ignition (C) CDI (B) breakers	Fuel Mixing (P) pre-mix (I) oil injection	Suspension (B) bogie (S) slider (P) pararail	Track length	Track width	Price $
Bombardier												
*A *B ELAN	129.2	250.4	1	1	F	M	B	P	B	289.6	38.1	1849
CITATION 3500	156.6	268.7	1	1	F	M	C	I	S	269.2	38.1	1979
CITATION 4500	169.3	368.3	2	1	F	M/E	C	I	S	289.6	38.1	2649
CITATION SS	172.0	368.3	2	2	F	M/E	C	I	S	289.6	38.1	2949
*B NORDIK	172.4	368.3	2	1	F	M	C	P	S	315.0	38.1	2849
*B SCANDIC	193.4	368.3	2	1	F	M	C	P	S	353.0	38.1	2949
*C EVEREST 500	208.2	496.7	2	1	F	M/E	C	I	S	315.0	41.9	3449
EVEREST L/C	229.6	462.8	2	1	L	E	C	I	S	315.0	41.9	3999
BLIZZARD 5500MX	190.9	496.7	2	2	F	M	C	I	S	289.6	41.9	3649

Table 3: cont.

BLIZZARD 9500	200.0	43.6.	2	2	L	M	C	I	S	289.6	41.9	4349
*A ALPINE	292.8	635.1	2	1	F	E	B	P	B	353.0	76.2	4899
Yamaha												
*D BR250F	128.0	246.0	1	1	F	M	C	I	S	238.0	38.0	1699
BT300F	157.0	294.0	2	1	F	M	C	I	S	259.0	38.0	2249
*C ET340F	165.0	338.0	2	1	F	M	C	I	S	275.0	38.0	2599
EC340F	173.0	338.0	2	1	F	M/E	C	I	S	275.0	38.0	2800
*B ET340TF	184.0	338.0	2	1	F	M	C	I	S	342.0	38.0	2899
SS440F	181.0	437.0	2	1	F	M	C	I	S	294.0	38.0	3499
EC540F	218.0	535.0	2	1	F	M/E	C	I	S	317.0	42.0	3599
SR540F	N/A	535.0	2	1	F	M	C	I	S	307.0	38.0	3999
SKX500F	N/A	499.0	2	2	L	M	C	I	S	207.0	38.0	4899
John Deere												
SPITFIRE	129	338	2	1	A	M/E	C	P	S	295	38	2399
TRAILFIRE	174	339	2	1	F	M/E	C	P	S	295	40	3699
TRAILFIRE LX	176	436	2	1	F	M/E	C	I	S	295	38	3699
SPORTFIRE	179	436	2	1	F	M/E	C	I	S	295	38	3899
LIQUIFIRE	187	436	2	2	L	M	C	I	S	295	38	4899

Table 3: cont.

Polaris												
TXL340	181	333	2	2	L	M	C	P	S	307	38	4339
TXLINDY 340	193	333	2	2	L	M	C	P	S	307	38	4769
CUTLASS SS440	172	432	2	1	F	M/E	C	I	S	279	38	3399
TXC 440	178	432	2	2	F	M/E	C	I	S	307	38	3989
TXLINDY 500	181	500	3	2	L	M	C	P	S	307	38	5449
Kawasaki												
INTRUDER 440	190	436	2	2	F	M	C	I	S	307	38	3699
LTD440	200	436	2	2	L	M/E	C	I	S	307	38	4499
INTERCEPTOR 540	200	530	2	2	L	E	C	I	S	295	38	5199
Arctic Cat	(1980/81	models)										
JAG4000	180	431	2	1	F	M/E	C	I	S	278	38	2899
PANTHER 5000	180	431	2	1	F	M/E	C	I	S	290	40	3199
TRAIL CAT	176	431	2	1	F	M/E	C	I	S	278	38	3399
PANTERA	177	500	2	1	F	M/E	C	I	S	278	38	3499
EL TIGRE 5000	174	500	2	2	A	M	C	P	S	278	38	3499

Table 3: cont.

EL TIGRE 6000	187	500	2	2	L	M	C	P	S	278	38	4199
Scorpion	(1980/81 models)											
TK340	169	339	2	1	F	M	B	P	P	284	38	2699
TK440	169	428	2	1	F		B	P	P	284	40	2699
WHIP	164	428	2	1	F	M	B	P	S	284	38	2599
STING 440	164	428	2	1	F	M	B	P	S	276	38	2799
SIDEWINDER	185	436	2	1	L	M	C	P	S	276	38	3999

ignition, oil injection, and track widths of 38cm. Not all of these features are detrimental to utility usage; in Chapter 4, greater detail will be provided upon the evolution of snowmobile design and the degree to which this has benefited recreational and utility usage respectively.

Figure 4 illustrates the distribution of model prices. It is noteworthy that, in the range below $2,500 choice is limited to only 5 of the total of 38 models made available for the 1981/82 season.

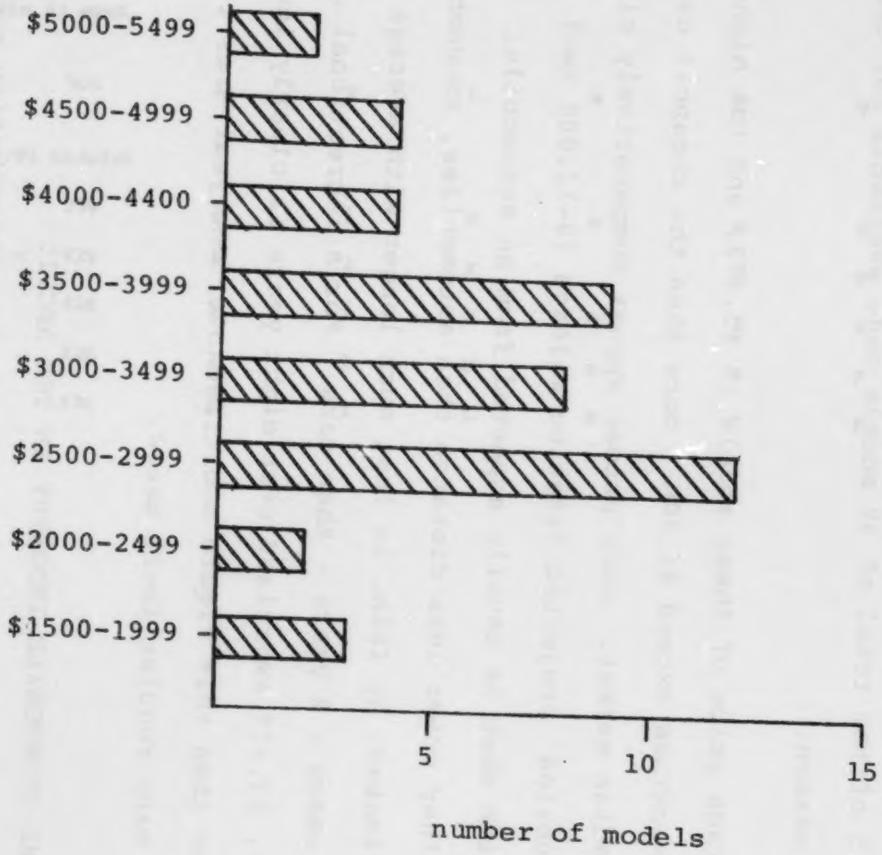
The average price of these models is $3,478, and the highest priced machines exceed $5,000 - more than the cheapest cars on the Canadian market. Such prices appear comparatively high when a snowmobiles' projected lifetime mileage (8-12,000 km) is compared with what is usually expected from an automobile. However, though they cover less distance than automobiles, snowmobiles are estimated, by ISIA, to last much longer with average recreational usage - 8 years - than cars. For a recreational snowmobiler, $3,478 amortized over eight years is clearly less of a burden than this figure amortized over one year - as it may be for many regular Inuit users.

2.3. THE SNOWMOBILE INDUSTRY IN THE ARCTIC

This section describes the major dealer distribution systems in the Arctic, typical facilities for service and repair and the situation regarding supplies of spare parts. Also, an approximation will be made of the size of the market in communities north of the tree-line. The section will conclude with

Figure 4

PRICE RANGES OF MODELS OFFERED 1981/82



a summary account of the efforts that manufacturers have made to discover and respond to the special requirements for Artic utility applications.

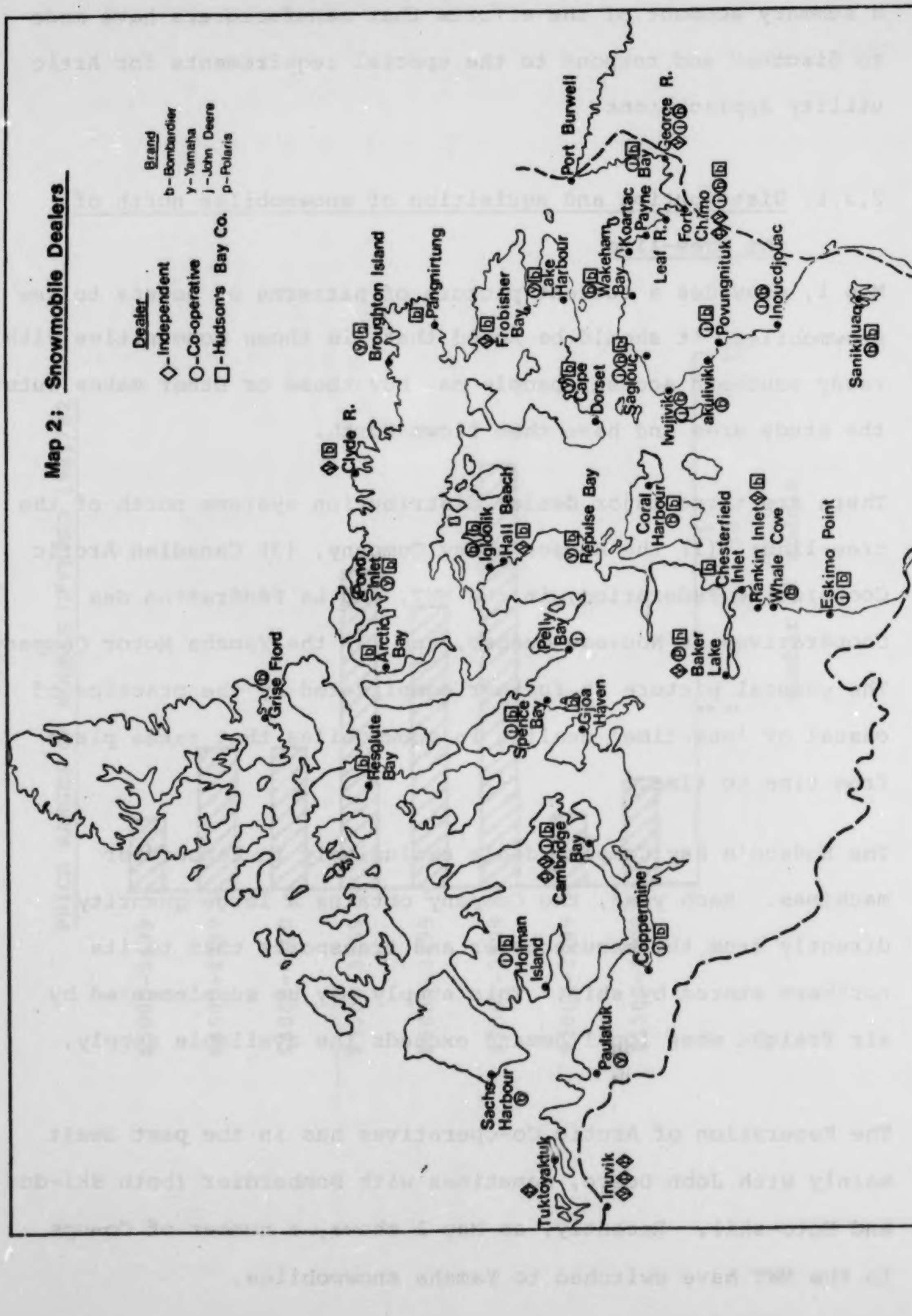
2.3.1. Distribution and aquisition of snowmobiles north of the tree-line

Map 1, provides a current picture of patterns of access to new snowmobiles. It should be noted that, in those communities with ready southern access, people may buy these or other makes outside the study area and have them flown north.

There are three major dealer/distribution systems north of the tree-line: (1) the Hudson's Bay Company, (2) Canadian Arctic Cooperative Federation, in the NWT, and La Fédération des Coopératives du Nouveau Québec, and (3) the Yamaha Motor Company. The general picture is further complicated by the practice of casual or 'one-time' dealing in snowmobiles that takes place from time to time.

The Hudson's Bay Company deals exclusively in Bombardier machines. Each year, the Company obtains a large quantity directly from the manufacturer and transports them to its northern stores by ship. This supply may be supplemented by air freight when local demand exceeds the available supply.

The Federation of Arctic Co-operatives has in the past dealt mainly with John Deere, sometimes with Bombardier (both Ski-doo and Moto-ski). Recently, as Map 1 shows, a number of Co-ops in the NWT have switched to Yamaha snowmobiles.


Map 2; Snowmobile Dealers
Dealer
◇-Independent
○-Co-operative
□-Hudson's Bay Co.
Brand
b-Bombardier
y-Yamaha
j-John Deere
p-Polaris
Tuktoyaktuk
Inuvik
Sachs Harbour
Paulatuk
Holman Island
Coppermine
Cambridge Bay
Resolute Bay
Grise Fiord
Spence Bay
Gjoa Haven
Arctic Bay
Pelly Bay
Baker Lake
Pond Inlet
Igloolik
Hall Beach
Repulse Bay
Chesterfield Inlet
Rankin Inlet
Whale Cove
Eskimo Point
Coral Harbour
Clyde R.
Broughton Island
Pangnirtung
Frobisher Bay
Lake Harbour
Cape Dorset
Sagluc
Ivujivik
akulivik
Povungnituk
Inoucdjouac
Sanikiluaq
Wakeham Bay
Koartac
Payne Bay
Leaf R.
Fort Chimo
George R.
Port Burwell

Over the last few years, the Yamaha Motor Company has established a dealer network north of the tree-line. This at first comprised four regional dealers: Inuvik, Cambridge Bay, Rankin Inlet, Frobisher Bay. The intention was for these dealers to supply those settlements linked by regional air lines. Buyers could then exercise the option of having their new machines flown in or they could drive them home. However, there are now twelve Yamaha dealerships. Of the 8 added to the original 4, 5 are Co-ops.

Other makes of snowmobiles than Bombardier, Yamaha and John Deere are less consistently represented. There appear to be no active Kawasaki dealers, Polaris dealers are placed in two locations, and Arctic Cat in two.

Occasionally someone in a settlement will obtain a batch of machines and dispose of them locally, thereby obtaining one or two 'free' machines. Also, in settlements with better southern connections, residents will often order their machines directly from the south and have them flown up. Yellowknife and Montreal are frequent sources for this trade. Some settlements and camps have no local access at all and must rely on dealers in other settlements.

Not surprisingly, the dealers able to take advantage of sea-lift for bringing snowmobiles north enjoy a distinct advantage over those obliged to rely upon air freight. When Willie Emudluk sells an Arctic Cat in George River, he must sacrifice almost 90% of the regular dealer profit in order to remain competitive.

The small local dealers interviewed in this study were all placed in a similar situation; lacking the capital to maintain an inventory and having to shave their margins to stay in business.

The larger and more extensive dealer networks possess a further advantage over the local independent dealers in that they have the resources to extend credit to customers. This, of course falls within the traditional northern practice whereby a trader would outfit a trapper to be later repaid in furs. Precise details of the extent of credit purchases could not easily be obtained during this survey. All that can be said is that John Deere operates a financing process through the Federated Co-op System, the Hudson's Bay Company extends credit on occasion, Yamaha dealers need not pay for their shipments until the January following sea-lift.

2.3.2. Facilities for service and repair and the supply of spare parts

Routine maintenance of snowmobiles, and many of the more common replacement or repair jobs, can be done without specialised tools and, in fact, most of those who use snowmobiles regularly for hunting are accustomed to carrying out their own repairs. One Bay store manager actually reported that his customers were not willing to entrust their machines to the mechanic who was on hand for work that may be required under warranty arrangements. Where special facilities or tools are required, the owners usually prefer to simply gain access to such facilities rather than have someone else operate them.

Dealers with whom this problem was discussed were fairly consistent in their view that, in the smaller settlements (that is most communities other than Frobisher Bay and Inuvik), there is simply not enough demand for a full-time snowmobile mechanic, even if he were to service all makes of machine. On the other hand, some kind of trained competence is required to deal with warranty claims.

Such claims must first be assessed, sometimes a delicate problem, and then dealt with. Sometimes the claimant will not want the mechanic/assessor to work on his machine but simply wishes to be given the replacement part - which, though it may save the dealer work, violates the proceedure for responding to warranty claims and may make him liable for further claims. While dealers who handle only snowmobiles will usually have someone available who can deal with such problems, general merchandising operations such as the Hudson Bay stores must go to special lengths to set up mechanics and facilities. The manager of one such store, which had a respectable sales volume, simply stated that the snowmobile side of his business was just not worth the effort when compared with the returns from his trade in other commodities.

Throughout the study area, independent snowmobile dealer/repair enterprises, not directly linked to a manufacturer or a merchandising system, were the exception rather than the rule and most of those encountered relied on some other source of income in addition to the income from their snowmobile business: running a local bus or taxi service, operating the local fuel franchise, or holding a job in agencies such as the Department of Public Works.

Several respondents expressed the view that, if the work related to all makes of snowmobile were to be combined with attending to outboard motors, three-wheelers and the occasional light truck, there might be sufficient work in most settlements to support a full-time fully trained mechanic.

Though this might be an agreeable situation, a detailed investigation of the prospects lies beyond the scope of this study. A more simple alternative might be the provision of facilities where snowmobiles and other machines could be worked on by their owners when their domestic facilities are inadequate. This has happened informally in some communities where the Hunters and Trappers Associations are fortunate enough to possess facilities which could be used as a workplace. Such facilities are sometimes adjacent to the office/garage of the local wildlife officer where, in one place or another, the owners of snowmobiles may occasionally be found working on their machines in the evenings.

A 'snowmobile problem' which is less susceptible to this kind of informal local resolution is the supply of spare parts, not so much those such as belts and plugs which are in fairly constant demand, but those parts which may be less regularly stocked, such as starter pulleys or pistons. Two years ago, at the Annual General Meeting of the Kitikmeot (Central Arctic) Inuit Association, Cambridge Bay, the Hunters' and Trappers'

Association described this as a serious problem, not so much in Cambridge Bay, which has a direct air link to Yellowknife, but in those central Arctic communities served by regional airlines through Cambridge Bay (see Map 1). The Hunters' and Trappers' Association recognised that this was principally an economic problem, the costs of maintaining a comprehensive spare parts inventory, and were at that time considering applying for funding to establish its own supply service.

In establishing its regional dealership, the Yamaha Motor Company addressed this problem by committing its dealers to have any part requested from an outlying settlement on the next flight by the regional airline, providing, of course that the part is stocked at the regional facility.

2.3.3. An approximation of the size of the market north of the tree-line

The following estimate should be treated with caution as there is no precise way of measuring how many snowmobiles are sold each season. Individuals may order machines from the south, parts of inventories may be held over unsold until next season, batches of machines may be flown from one settlement to another and recorded twice. However, a general and qualified approximation is worth attempting since so many of the manufacturers' representatives interviewed in this study stated that the size of the Arctic market is critical in determining how much effort manufacturers will spend to meet its demands.

The estimate attempted here extends over the NWT and Northern Quebec. As a starting point, the ISIA records of shipments to the Territories show an average of approximately 2,000 units per year over the five seasons, 1976-81. We were provided with figures for shipments by the Yamaha Motor Company to their northern dealers, but unfortunately directly comparable data from other manufacturers to dealers could not be obtained as these transactions are either mediated through major distributers such as the cooperative federations or the Hudson's Bay Company, or made on a local/regional opportunity basis. However, the Fédération des Coopératives du Nouveau Québec provided us with similar data, which concerned Bombardier and John Deere Snowmobiles. Cooperatives in the Territories were all called individually, with the results showing a mix of all currently available makes of snowmobiles. For data on Bombardier machines sold through the Hudson's Bay Company, all the stores were contacted. Some managers were freshly appointed and unfamiliar with the local market or the sales volumes of their predecessors; a few were disinclined to provide information. Some sources were kind enough to provide the information required but requested that full details not be revealed unless precisely similar data is also tabulated for their competitors. This consistency could not be achieved under the survey circumstances and therefore the data provided below is listed without reference to specific makes or models.

The figures presented in Table 4 are, like those in Table 1, for snowmobiles shipped into the communities. We did obtain figures

Table 4: COMPILATION OF DATA ON SNOWMOBILE SHIPMENTS AND THEIR RELATION TO POPULATION

SETTLEMENT	TOTAL SHIPPED	LOCAL POPULATION	NUMBER OF INHABITANTS PER SNOWMOBILE	COMMENTS
N.W.T.				
Igloolik	32	843	26	new management data uncertain
Pond Inlet	53	708	13	-
Resolute Bay	6	206	34	some ordered elsewhere
Lake Harbour	32	306	10	-
Clyde River	44	471	11	-
Broughton Is.	21	398	19	-
Arctic Bay	7	474	67	uncertain data
Grise Fiord	2	110	67	machines ordered elsewhere
Hall Beach	4	400	100	machines bought in Igloolik
Pangnirtung	72	995	14	-
Frobisher Bay	69	3255	47	-
Sanikiluaq	28	366	13	-
Rankin Inlet	108	1289	12	some sold elsewhere
Baker Lake	115	1163	10	-
Chesterfield In.	7	292	42	some bought in Rankin Inlet
Coral Harbour	51	532	10	-
Eskimo Point	41	1019	25	some bought in Churchill
Repulse Bay	34	338	10	-
Whale Cove	4	208	52	some bought in Rankin Inlet
Cape Dorset	79	782	10	-

Table 4: cont.

Cambridge Bay	108	969	9	some sold elsewhere
Coppermine	69	916	13	-
Holman Is.	29	349	12	-
Pelly Bay	1	295	295	most bought Cambridge Bay
Spence Bay	36	536	15	some ordered from Cambridge Bay
Gjoa Haven	28	519	18	-
Tuktoyaktuk	N/A	-	-	-
Paulatuk	15	183	12	some ordered elsewhere
Sachs Harbour	7	197	28	some ordered elsewhere
N.W.T. Sub-Totals	1,144	18,118	16	
Québec				
Great Whale	20	644	32	some bought elsewhere
Inoucdjouac	33	745	22	-
Fort George	0	69	-	ordered elsewhere
Povungnituk	41	704	17	-
Akulivik	26	188	7	-
Fort Chimo	138	976	7	some sold elsewhere
Leaf Bay	0	102	-	ordered elsewhere
George River	30	315	10	-
Payne Bay	30	277	9	-
Port Burwell	0	70	-	ordered elsewhere
Koartac	0	136	-	ordered elsewhere
Wakeham Bay	24	273	11	-

Table 4: cont.

Ivugivik	20	179	9	-
Sugluk	40	495	12	-
Québec Sub-Totals	402	5,212	13	
TOTAL	1,546	23,330	15	

for units sold so far and for those still on inventory, but, in the month, February, when the survey was made, most dealers fully expected sales to pick up with the onset of Spring.

For each community, the population estimates are listed. For the NWT, these are extrapolated from Hamelin's (1979) projections for population increases between 1978 and 1986. For Northern Quebec, the figures were provided by Makivik Corporation.

In calculating the relationship between snowmobile shipments and local populations, the intention was to see if any consistent ratio would appear, which could be applied to estimating demand on the basis of population. This ratio varies widely bewteen settlements. Where local snowmobile sales were low in relation to population, respondents either refered this to lack of local outlets (e.g. Pelly Bay, Grise Fiord) or to the fact that local residents were accustomed to acquiring their snowmobiles from dealers in nearby settlements (e.g. Whale Cove from Rankin Inlet, Hall Beach from Igloolik, Sachs Harbour from Inuvik). Where sales are high, the presumption may be that the settlement acts as a source of snowmobiles for Inuit residing elsewhere (e.g. Cambridge Bay).

However, when shipments to all the NWT settlements returning data are compared with the population in those settlements, the ratio of one snowmobile for every 16 persons is obtained. In Northern Quebec, where the data on sales and shipments was more consistent and reliable, the comparable ratio is one snowmobile for every 15 persons.

If snowmobile shipments are compared with the number of Inuit men of working age (15-65) projected for the NWT, for settlements listed, the ratio is approximately 1 machine for each 4 men (the comparable ratio in the age group 20-60 years is 1:3).

For the 1976, Inuit Land Use and Occupancy Study, Freeman (1976) estimated the total number of "adult Inuit hunters" in the NWT to be in the neighbourhood of 2,000. The total Inuit population increase 1976-81 has been estimated at 17% (Hamelin, 1979). If this figure is increased by this percentage, the resulting ratio is one machine shipped for every two hunters. The comparable figure for the machine/hunter ratio in Northern Quebec is 402/588, or 1:1.5.

These figures are deliberately kept general and conservative. They do not include snowmobiles ordered from the south by individuals, or fetched from locations outside the study area, such as Churchill.

Nevertheless, there is consistency enough in the data to support two conclusions. The first is that the snowmobile market in the study area is in the order of 1500-1600 units. Second, the ratio of new machines to the number of people regarded by authoritative sources as hunters is high and suggests an extraordinarily high rate of turnover.

2.3.4. Manufacturers' efforts to discover and respond to the requirements of the Arctic utility market

All the manufacturers currently producing snowmobiles were asked about their perspective on the Arctic utility market, whether or not they had made particular efforts to discover its special requirements and to respond to them by modifying existing machines or designing new models. With one exception, all the manufacturers had made such efforts in recent years. Kawasaki's position* was that the company is currently committed to a limited annual production of 5,000 units and at this level economics of scale would not permit them to produce a competitive utility machine. The efforts undertaken by other manufacturers are described below: overall, the results support the view that producing an Arctic utility snowmobile is an economic rather than a technical problem.

In 1979, production engineers from Polaris visited several Alaskan communities to find out what kinds of features users felt should be incorporated in a snowmobile suitable for hunting, fishing and trapping. Using the information gained, the company constructed two prototype models based upon two current models, the Cutlass and the Galaxy (see Photos 1 to 6). The prototypes had elongated tracks, reinforced bodywork, enlarged windshields, heavy duty towing hitches, extra wheels supporting the tracks, pusher plates at the front end, kick stands to jack the track off the snow, reduced gearing, larger fuel tanks and standard 440 twin-cylinder engines.

The prototypes were displayed at the 1980 dealer convention in Winnipeg. However, in spite of enthusiasm over the machines as utility vehicles, the projected prices were such that not enough dealers were willing to place the volume of orders required to

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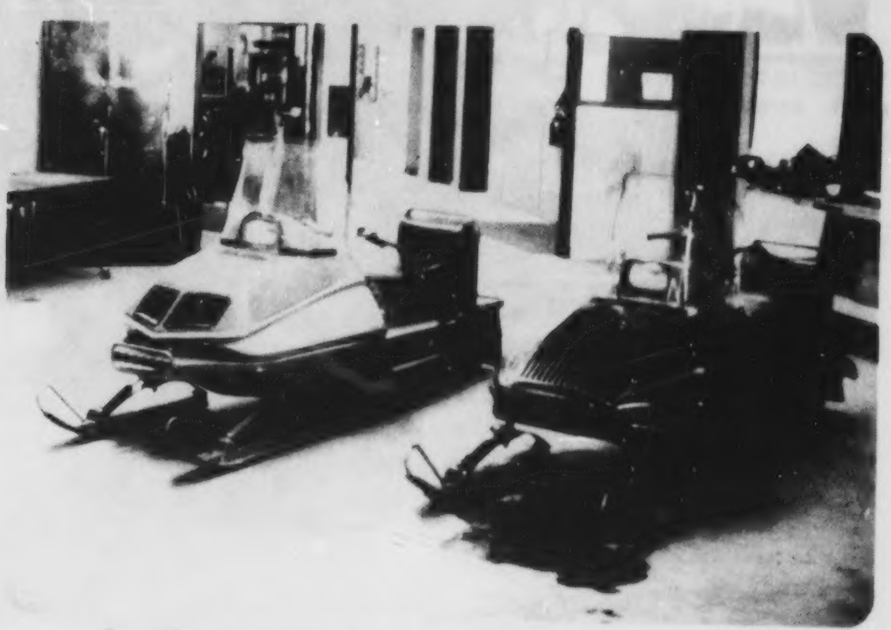
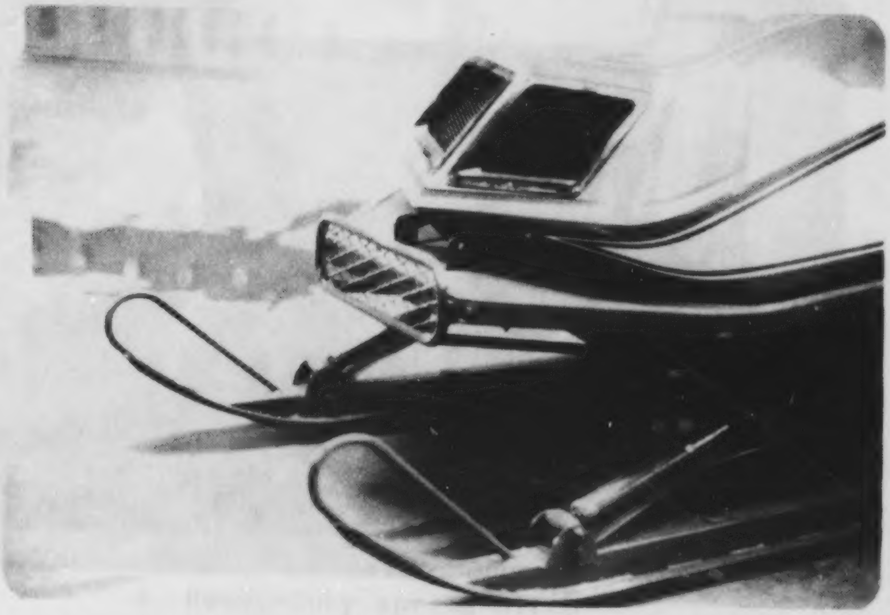
* Kawasaki has since withdrawn from all snowmobile production.

Photos 1-6

DETAILS OF UTILITY PROTOTYPES BUILT BY POLARIS AFTER USER RESEARCH IN ALASKA

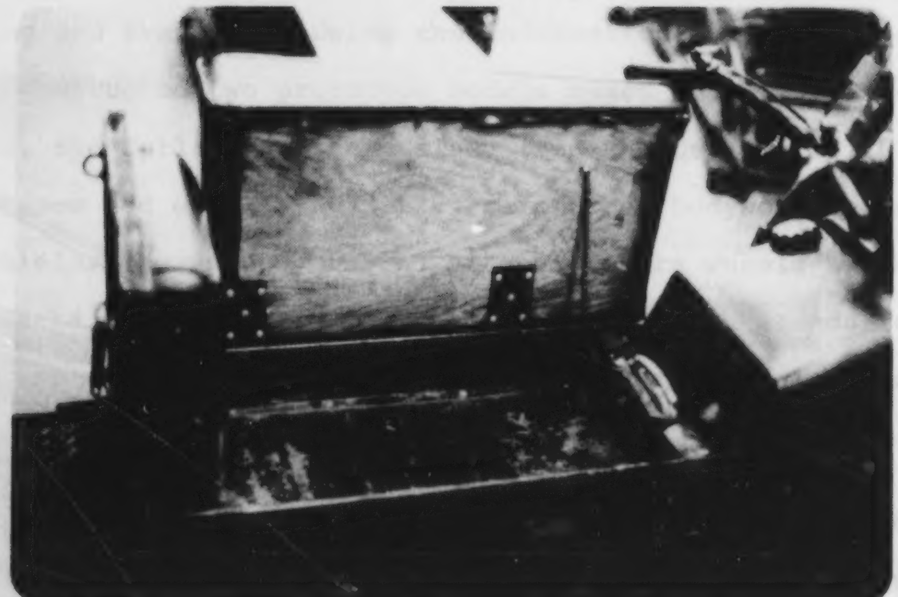


1. The two prototypes

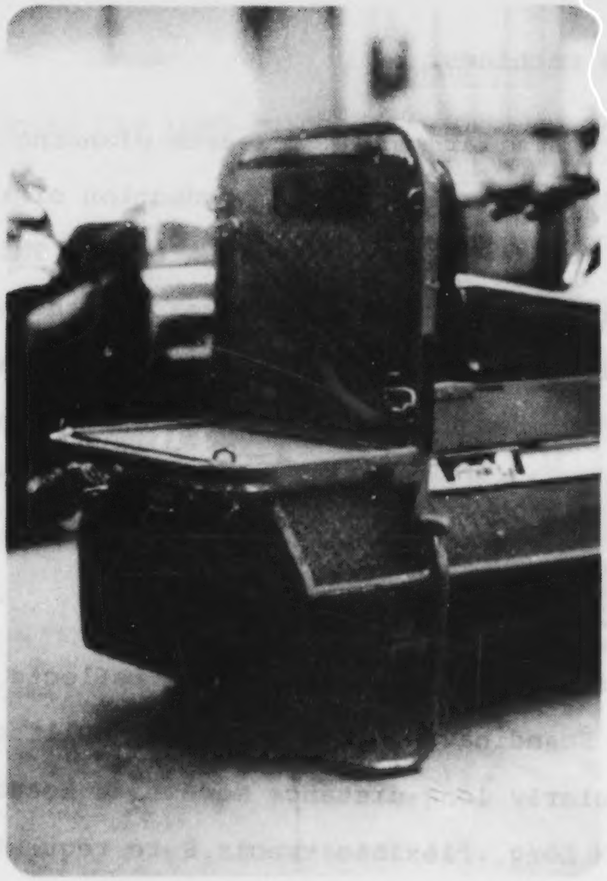


2. Reinforced skis, front & pusher plate

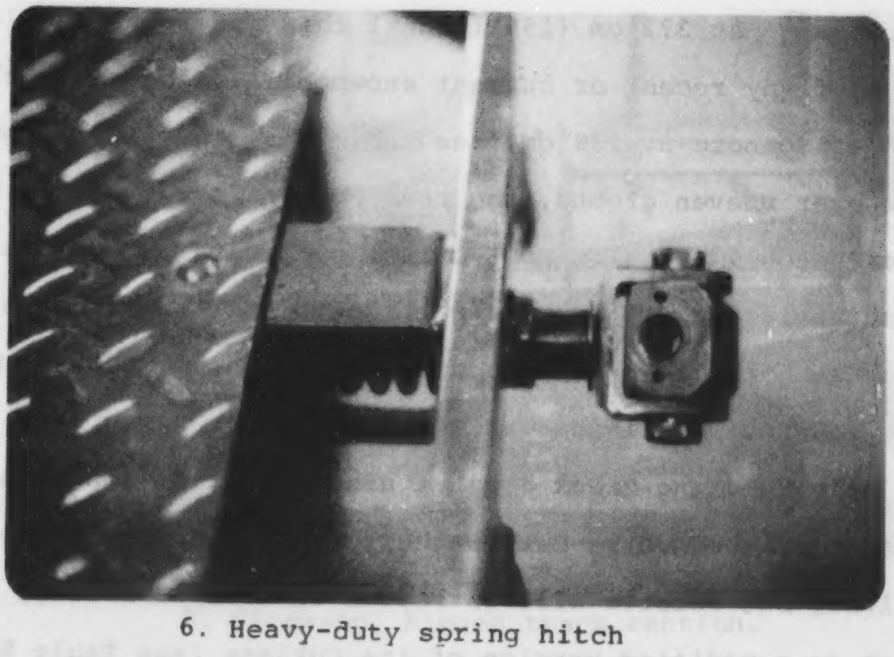
3. High, wide, windshield



4. Extra baggage area



5. reinforced rear end, baggage area, kickstand.



6. Heavy-duty spring hitch

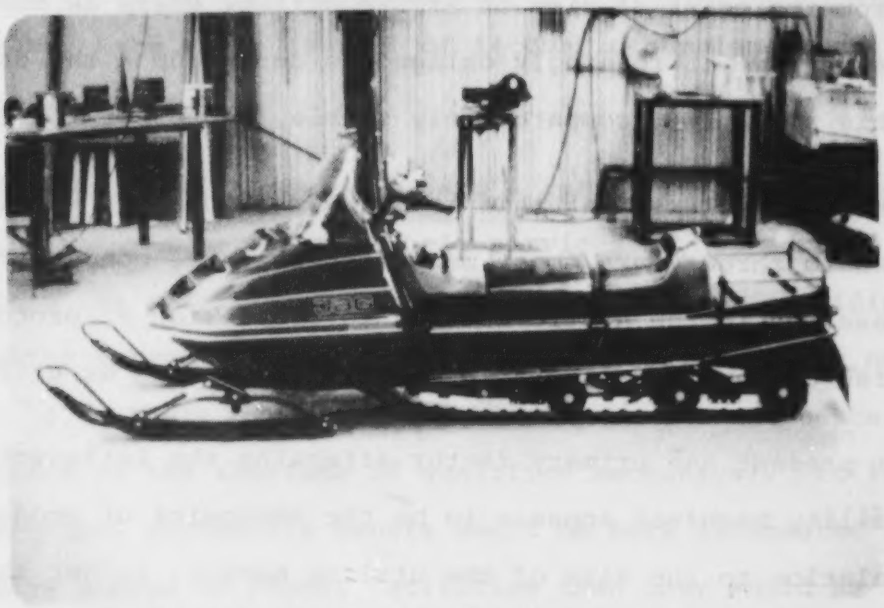
justify producing the machines.

John Deere undertook a similar market research planning exercise in the Canadian Arctic. This led to the production of the Arctic Trailfire Special, a modification of the Trailfire. This model is distinguished by extra wheels to reduce wear on the sliders, adjustable roller idler on the chain, protected chain case, fuel primer pump, enlarged windshield, on/off light switch, heavy duty hitch, steel instead of aluminium steering components, reinforced fron suspension, fuel shut-off valve and a standard 340 twin-cylinder engine.

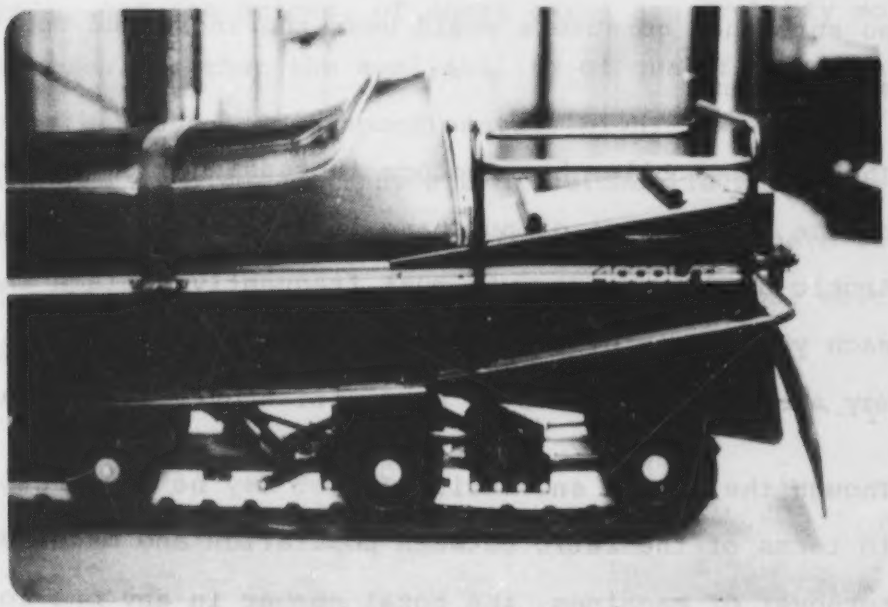
Arctic Cat's production of utility snowmobiles reflects that company's experience in Scandinavia. For that use, pulling is also at a premium, particularly long-distance movements keeping pace with reindeer herds. Long, flexible tracks were requested, and the company produced a special version of its Jag, the Long-Track Jag. At 372 cm (155 inches) this model has the longest track of any recent or current snowmobile (closest is the Bombardier Scandic at 239 cm, see Table 3). To maintain optimum contact over uneven ground, the rear part of the track is hinged (see Photos 7 and 8). Most of the run of this model was sold in Scandinavia, but a few remain in North America. Though the company has formally ceased production, at the time of compiling this report, Polaris has acquired the assets and plans to produce not only the Long-Track Jag but also a similar long-track version of the Arctic Cat Panther. This plan may now be shelved, but Polaris will introduce in 1982/83 a long, hinged track on a modified version of its Cutlass (see Table 9).

Photos 7-8

ARCTIC CAT LONG-TRACK JAG - DEVELOPED FOR WORKING USE IN SCANDINAVIA



7. Long-track Jag, side view.



8. Close-up, hinged track section.

2.3.5. The snowmobile industry - concluding remarks

The decline in the recreational snowmobile market, which as yet snows no sign of reversing itself, may turn out to be beneficial from the point of view of Arctic utility users as manufacturers may become increasingly obliged to depend upon the demands of this small, yet comparatively stable, sector of the total market.

There are signs that this is already happening. Over the last two to three years, most manufacturers have conducted market research in the Arctic leading to the design of prototypes, modified current models or new models, advertised as utility machines.

At present the primary factor affecting the introduction of utility machines appears to be the economics of production in relation to the size of the utility market, rather than technical innovation; the manufacturers feel that they know how to produce tougher, more utilitarian machines but they are not so sure that consumers would bear the increased costs of such snowmobiles.

Though snowmobiles have become increasingly expensive, the effect of the intial cost of purchase is felt far more severely by Arctic utility users, who must frequently replace their machines each year, than by the average southern recreational user who may amortize the costs of acquisition over eight years.

Though the Arctic snowmobile market may be comparatively dense, in terms of the ratio between population and machines and the turnover of machines, the total number in any one location is not usually sufficient to support snowmobile dealing as an independent business. Snowmobile dealing is either conducted along with general merchandising, as with the Hudson's Bay Company, or is integrated with informal local economies, whereby a dealer may have other sources of income to supplement that from snowmobile sales.

Qualified snowmobile mechanics are a rarity in the study area, but, in fact the demand for their services may be more for assessing the validity of warranty claims and for providing and performing warranty work, than for other repairs. Most owners prefer to do their own repairs (this may be a consequence of experience of the shortage of qualified mechanics). It is conceivable that snowmobile owners would be more interested in obtaining access to repair facilities than they would be to qualified mechanics.

The supply, and the prices, of spare parts are probably more critical problems than the availability of qualified mechanics. The prices can be as disproportionately high, in relation to their cost on a new machine, as they are on automobile and, however, high the prices, it is not unusual for an owner to have to wait several more months for a spare part other than those which are stocked as a matter of course.

3. INUIT USAGE

> When they started getting skidoos, people decided that they could not keep their dogs going. When you try to keep dogs, the policeman just shoots that, so we turn to metal things that cannot be shot. You break them up yourself, or bump somebody with them . . . . By using a skidoo as your means of transportation, you tend to sharpen your own senses: smell, eyes, your sense of direction in the dark or in a storm, and your prediction of weather. These things were not really sharp before, because you had dogs. If you were going to go somewhere, they would probably know which way to go anyway, because all the things you knew went to the leader and he recorded where he had been before. Dogs used to be valuable for ice. They learnt to walk on different kinds of ice. Today, you cannot get a skidoo to walk on different kinds of ice. You have to try and analyse that ice yourself, before you go on it with the skidoo. That way you sharpen your sense - senses that were not so sharp before.
>
> William Kalleo, Nain
> Brice-Bennett, 1977:323

3.1. TRACTIVE PERFORMANCE

The most consistent distinction between Inuit use of snowmobiles and southern recreational use is that the machines should be capable of pulling heavy sleds for considerable distances, over far roughter terrain than the groomed trails for which the majority of modern machines are designed. This was stated to be the basic requirement; for the snowmobile to be useful as a tractor. Further to this, there were more specific purposes which enabled a rough categorisation to be made of three major machine/use types.

The first category is of machines capable of pulling loads of a ton or more at steady speeds over long trap-lines. The machine's suitability for other purposes is secondary. At present, the Bombardier Alpine is the only machine capable of providing this level of performance.

The second kind of use respondents refered to was for a small, maneuverable, machine which would handle well in deep snow, in hilly terrain and amongst trees. The Bombardier Elan is widely perceived to fulfill these purposes best, though some Elan owners allowed that they would prefer the Alpine if they could afford the extra (triple) cost.

At present, these two categories are more or less dominated by these two models. They have both been on the market for over ten years and owners tended to identify the pattern of use with the machine to which they had become accustomed. In the small machine category, there is now competition to the Elan; how effective this will be remains to be seen.

The third category is the market segment at which the range of machines discussed in subsection 2.3.4. is aimed. The requirement here is for a machine capable of pulling around 450 kg (1000 lbs.) yet also capable of a fair turn of speed - approaching 100 km/h (63 mph). The extent to which such speeds are reconcilable with such lugging properties is perhaps the most difficult design problem in producing such an all-round kind of machine.

A load of 450 kg (1,000 lbs.) was the most common figure advanced as a minimal capability. Muller-Wille (1978) found that Repulse Bay hunters were used to pulling the same weight in the early 1970's, and Usher (1972) in noting that the Bankslanders also pulled such loads, observed that 450 kg (1,000 lbs.) was about the payload that could be pulled by a full team of 9 dogs.

Payloads of this magnitude are considerably higher than those for which snowmobiles are designed. Even the Bombardier Alpine, when tested for military purposes, was limited to a load of about 200 kg (450 lbs.). Manufacturers with whom the question of tractive power was discussed were well aware of the loads that are customarily pulled in the course of Arctic use, and are willing to provide for this by strengthening hitches and lowering gear ratios, but they are disinclined to advertise specific figures for tractive capacities on account of warranty problems that might arise.

3.2 VARIETIES OF USE

Of the various uses to which snowmobiles are put in the course of hunting, trapping and fishing, trapping is perhaps the least demanding in the sense that the trapper will be proceeding at a steady and moderate pace over a regular route during a period in the winter when the snow cover is the most extensive. On the other hand, trapping may account for greater distance covered than other forms of use.

At various times throughout the winter, people may fish with nets below sea and lake ice. In many places, a typical weekend trip will be to and from a fishing camp which may be well over 160 km (100 mi.) from the settlement.

Hunting for caribou, musk oxen or wolf may entail journeys over considerably distances and over wider varieties of terrain than would be encountered in journeys over regular traplines or to known fishing places.

Hunting whales, seals, walruses and polar bears necessitates considerable travel over sea ice, probably the least avoidable source of wear and tear on snowmobiles. This problem is accentuated in the spring when the machines must run directly on the ice and through the melt-water that has collected on the surface.

It is during the fall and the spring seasons that terrain conditions are at their worst. During the fall, the snow may be late, the cover discontinuous, and travellers may encounter considerable stretches of bare ground, rocky outcrops or vegetated surfaces. During the spring, such conditions reappear and are compounded by the increasing softness of the snow layer and the mud and moisture produced by the thaw.

On the sea ice during this season, hunters must contend not only with a surface of rough and broken ice mixed with slush and melt-water but also with the problems created by the break-up of the landfast ice. Travel at this season may entail jumping snowmobiles and sleds across open leads of up to 2.4 m (8 ft.) in width (this being given as the everyday maximum). This is one use for which the higher speeds are appreciated.

3.3. SOURCES OF OPERATIONAL STRESS

When asked about the typical kind of repair and replacement problems that might arise, even with the better designed multi-purpose machines, mechanics and dealers provided responses which fell roughly into three categories: (1) Problems arising from improper care and maintenance; (2) those referable to extreme operating conditions; and (3) those due to the machines being stressed beyond their limits, even well-maintained machines.

The majority of mechanics and dealers consulted were consistent in the view that, when a problem could not be traced to inherent weakness in design or materials, the primary cause of mechanical breakdown or malfunction would be improper care and maintenance by the owners. Frequent examples of such negligence were: the failure to warm engines sufficiently during very cold spells, inattention to the adjustments that should be regularly made to the tension on belts and tracks, failure to take the precautions necessary when travelling conditions were such that sliders would not receive sufficient lubrication, failure to notice or replace broken bogie springs. While making such comments, it was clear that dealers and mechanics had certain of their customers in mind and the remarks were invariably evidenced by references to specific cases where an identical part of model performed well when properly cared for.

From these discussions a distinction emerged between commonplace negligence (e.g. failure to warm engines sufficiently) and lack of a certain sensitivity to slight signals of performance falling off or parts moving out of adjustment. One mechanic felt that operators should be able to tell from the slight changes in engine sounds when the oil/gasoline ratio in oil injection systems was altering due to extreme cold (see subsection 4.2.3.).

Similarly, problems arising from changes in track tension, or alignment between tracks and sliders, should be detectable in the course of operating the snowmobile. Such problems as these were referable ultimately to individual interests and aptitudes. One mechanic, responsible for servicing a number of machines used by government employees, was confident that he could predict the problems that would arise from this person or that taking out a machine for a few days.

The second source of operating stress is one over which the user has little effective control. A major example of this is the effect that extremely low temperatures will have upon the materials used to fabricate snowmobiles. Also, however well-maintained a machine may be, to set it in motion at all when temperatures are in the area of minus 40° C is to induce stress beyond the levels that snowmobiles are built to withstand.

The third source of stress is one over which opinions differed about whether it was simply a version of one of the two above. Even the 'best' snow conditions north of the tree-line are more severe than those for which recreational snowmobiles are designed. Muller-Wille (1978) points out that the wind-packed snow surfaces typical of the Canadian tundra are substantially more damaging to snowmobiles than their counterparts in the Scandinavian Arctic. To these, normal, conditions may be added marginal conditions that will be encountered throughout the winter, but specifically during the fall and spring seasons: exposed surfaces of gravel, rock and vegetative mats, rough, bare, ice surfaces, melt-water and mud.

Opinions differed over the extent to which using a snowmobile in such conditions is simply an example of the first of the above categories - improper care and maintenance. Conversely, there was the view that, collectively, Arctic operating conditions conformed to the second category and, like the extreme cold, were facts of life that had to be lived with and their effects alleviated as far as possible. Thus, an owner would see little contradiction between maintaining his machine with great care but at the same time, when circumstances call for it, subjecting the machine to stresses well beyond those contemplated by designers. Thus, one trapper in the Mackenzie Delta, who had always relied upon the Bombardier utility model, the Elan, would use it to pull loads of 450 kg (1,000 lbs.) over hilly terrain, and would expect to replace the clutch several times throughout the winter; the trade-off was worth it to him.

It should be pointed out here that manufacturers do not advertise design limits, either with respect to tractive power, or the terrain conditions. On some models, it is stipulated that the warranty will not apply if a passenger is carried. Such a commitment to specific capabilities, such as tractive power, could place manufacturers in an equivocal position regarding warranty responsibilities. If, in recognition of the fact that Inuit regularly pull 450 kg (1,000 lbs.) with 340/440 models, the manufacturers were to advertise that as a performance feature, they would be required to support this by warranty commitments which could precipitate a rash of claims from southern users who might elect to try pulling such payloads around the countryside. On the other hand, if the towing capabilities implied in the stock sleds become the basis of an explicit commitment upon performance, and these are in the 100-150 kg (220-330 lbs.) range, then the implication would follow that those Inuit who elected to pull larger payloads would be automatically disqualified from warranty claims. The manufacturers with whom this was discussed expressed pleasure at the demonstrated tractive power of their snowmobiles, and expressed a willingness to install stronger hitches and reduce gear ratios so as to improve this further, but are understandably reluctant to make any specific claims about such capabilities.

A similar situation obtains with respect to snowmobile performance over 'marginal' terrains, such as the mosaic of bare, ridged, ice, slush and melt-water characteristic of the spring thaw. Such conditions are so pervasive, and their objective categorisation would be so complex, that an attempt to delimit performance capabilities with respect to terrain conditions would be counter-productive in its impracticability. In its absence, the only resort is to subjective assessment and this of course leads to frequent local dispute over the validity of warranty claims (see subsection 2.3.2.).

To some extent, such present problems are a product of the history of snowmobile development. Early models were widely perceived to be examples of all-terrain-vehicles (see Annex 1) and the comparative versitility of current bogie-suspended models provides a contrast with the recently evolved slider-suspensions, which undeniably provide greater speed and comfort but, on a more limited range of terrain types.

This question will be taken up again towards the end of section 4.6 - after a review of comments upon snowmobile performance under Arctic conditions. Before this, in the following two sections, an attempt will be made to estimate both the annual costs of snowmobile operation and the extent to which these may rise over the next 5 years.

3.4. LONGEVITY OF SNOWMOBILES IN THE ARCTIC

When asked how long a snowmobile lasted when used regularly for hunting, trapping and fishing, the common reply from Inuit users was one year. This would not necessarily mean that the machine would be totally unuseable after one such year; it would more often mean that, after a season of serious use, the probability of breakdown during the second year would rise to the point where a user would go to some lengths to obtain another machine. His machine of the previous year would not necessarily be scrapped; it may be retained as a spare, passed to another member of the family, restricted to local use, or sold in the community. When the means to buy a new vehicle are not available, the owner has little choice other than to overhaul the machine and replace critical parts.

The estimate of one year of serious and reliable use was not confined to those who use recreational models for utility purposes. Even the highly successful trappers from Sachs Harbour, who can afford to purchase Bombardier Alpines ($5,000) will obtain a fresh machine each year whenever possible. A breakdown on the trap-line may result in several days of trapping lost if a spare is not carried. The need to regularly attend to and adjust an old machine may mean spoiling the trap sets with traces of the smell of oil and grease. The Alpine is valued amongst Banks Island trappers for its capability in towing a payload of a ton, enabling the trappers to make longer runs or take larger complements of supplies, spare parts and fuel than that possible with the mid-size machines.

Owners, mechanics and dealers were frequently asked what kind of distance would be covered throughout a year of active use. Estimates given were surprisingly higher than the few, earlier, recorded examples. Usher (1972) estimated that the snowmobiles in use during the period, 1967-70, would last an average of 2 years and would cover about 1,600 km (1,000 mi.) during each year. Muller-Wille (1978) provided a slightly higher estimate for Repulse Bay in 1973: he estimated that snowmobiles would last for one year on an average and cover 2,500 km (1,562 mi.) during that period.

Both of these estimates were calculated from detailed information gathered over some time in both these settlements. For this survey, respondents were simply asked what kinds of distances they associated with 'serious' snowmobile use. Answers were provided with qualifications regarding condition, economic and environmental, but though they encompassed a wide range, 8,000 km (5,000 mi.) to as much as 16,000 km (10,000 mi.), they were invariably in excess of the estimates worked out for the earlier periods mentioned above.

There could be several reasons for such an increase. Snowmobiles have become faster and, perhaps more importantly, faster in more comfort. Those respondents who were grateful for the advance from bogie to slider suspensions, would point out that not only can current machines go faster but they can go faster for longer with less wear and tear upon the driver (see section 4.4.).

A further factor may be the difference in use. Usher (1972) estimated that in the period 1967-70, trappers would cover an average of 32.5 km (20.3 mi.) per day on the trapline. By comparison, it would not be unusual for a hunter in the eastern Arctic to travel 100-150 km (63-94 mi.) to and from a regular weekend hunting or fishing camp, and while there, he may put more mileage on his machine. While on such journeys, he may be exceeding the daily distance estimated above during each hour.

What is of perhaps greater significance than this estimated increase in annual distances covered is that, while his has happened, the life-spans of snowmobiles have not changed to any appreciable degree. These two studies revealed that snowmobiles during this period, 1967-73, would last one or two years; that is still the general case, but during these periods modern machines would cover appreciably greater distances than the older models.

Data from two other sources can be compared with the estimate of 8,000-16,000 km (5,000-10,000 mi.) as a crude estimate of the lifetime range of a modern snowmobile. The manufacturers abide by the estimate of longevity for recreational snowmobiles provided by ISIA, which is 8 years - and is based upon an analysis of snowmobile registrations in the United States. Asked what they considered to be the annual distance covered by recreational users, manufacturers replied in the range of 1,440-2,240 km (900-1,400 mi.). (Some manufacturers also observed that annual recreational mileage had increased considerably with the advent of the recent more comfortable, quieter and faster machines.) These estimates yielded a projected lifetime mileage for recreational use of 11,520-17,920 km (7,200-11,200 mi.).

Hydro-Quebec use Bombardier Alpine snowmobiles for monitoring lines across country. For this use, a machine will last two years, after which it is scrapped or retired for use around the depots. During these two years, the machines will be driven for an average total of 500 hours. Military tests of this model (L.E.T.E. 1964) yielded cross-country speeds of 5.6-8.4 km/h (9-13.5 mph.) which suggest that the distance likely to be covered during this 500 hours of cross-country work would be in the range of 7,200-10,800 km (4,500-6,750 mi.). (The military data was needed for these rough conversions because even Hydro-Quebec did not keep accurate records of distances covered.)

In the following section, an attempt will be made to estimate the costs of snowmobile operation and to provide a perspective upon how these costs are likely to change over the next five years.

3.5 SNOWMOBILE OPERATING COSTS: CURRENT AND PROJECTED TO 1986

The estimates presented in Tables 5, 6, 7 are based upon the following limits and assumptions:

Purchase Price: Three sizes of snowmobile are entered in the tables, with purchase prices reflecting those that Arctic buyers are faced with. Prices vary widely with the region and the dealer network. Those used here are conservative. The $1,700 refers to small machines such as the Bombardier Elan, the $3,300 refers to the mid-size dual purpose machines such as the Yamaha 340TF and the John Deere Arctic Trailfire Special, the $5,000 to the class occupied by the Bombardier Alpine. An annual price increase of 10% is assumed; again, conservative by comparison with increases in recent years.

Fuel Prices: At present, in NWT communities north of the tree-line, these range from a low of 53¢/litre (Repulse Bay) to a high of 75¢/litre (Pelly Bay). The average (including a new tax shortly to be imposed is 58¢/litre. The Petroleum Products Division of the NWT government expect that prices in the settlements will double by 1986. Therefore, for the purposes of this projection, this increase will be assumed according to the following increments:

1981/82	58¢/litre
1982/83	72¢/litre
1983/84	87¢/litre
1984/5	101¢/litre
1985/6	116¢/litre

Costs of Spares and Overhauls: The designation "spares" covers the parts that would be required over a season of operation without major mishap and includes items such as plugs, belts, minor structural replacement parts and possibly sliders. The costs for overhaul refer to the steps an owner would need to take in order to re-condition a machine for a second winter of use. The major item is the track (cost, $400-$500). Other possible components would include, new skis, suspension elements, engine overhaul. The costs for an overhaul at current rates are estimated at between $700 and $900. These are again conservative; many users estimate that the cost of re-fit is about half the price of a new machine. These are also projected to rise by 10% p.a.

Distances Covered Each Season: Costs are estimated for two distances. The minimum distance respondents gave for 'serious' use approximated 6,000 km (3,750 mi.). The maximum was in the range 12,000 km (7,500 mi.) - though a few respondents gave yet higher figures. (Though covering the longer distance would entail yet further expenditures on spares, this extra cost is ommitted for the sake of simplicity).

Fuel Efficiency: These are based upon estimates provided by users. For the small machines, 7 km/litre (20 mpg) was felt to be a good working estimate when pulling a load. Without a load, these machines may be considerably more fuel efficient. For mid-size machines 5 km/litre (14 mpg) while pulling full loads (respondents provided wider variations in their estimates for these machines, primarily, it seems, because there are more models and engine types involved). For the Bombardier Alpine 3 km/litre (8 mpg). Thus figure tallies with the estimates of Hydro-Quebec and the results of military test.

Table 5: SAMPLE OPERATING COSTS FOR A SMALL SNOWMOBILE

	81/82	82/83	83/84	84/85	85/86
First Year					
Purchase	1,700	1,870	2,057	2,263	2,490
Spares	200	220	242	266	293
Fuel/6,000 km	497	617	746	866	994
TOTAL	2,397	2,707	3,045	3,395	3,777
¢/km	0.40	0.45	0.51	0.57	0.63
Fuel/12,000 km	994	1,234	1,492	1,732	1,988
TOTAL	2,894	3,324	3,791	4,261	4,771
¢/km	0.24	0.28	0.32	0.36	0.40
Second Year					
Overhaul	700	770	847	932	1,025
Spares	200	220	242	266	292
Fuel/6,000 km	497	617	746	866	994
TOTAL	1,397	1,607	1,835	2,064	2,312
¢/km	0.23	0.27	0.31	0.34	0.38
Fuel/12,000 km	994	1,234	1,492	1,732	1,988
TOTAL	1,894	2,224	2,581	2,930	3,306
¢/km	0.16	0.18	0.21	0.24	0.27
2 yr total/ 12,000 km	3,794	4,314	4,880	5,459	6,089
¢/km	0.32	0.36	0.41	0.45	0.51
2 yr total/ 24,000 km	4,788	5,548	6,372	7,191	8,077
¢/km	0.20	0.23	0.27	0.30	0.34

Table 6: SAMPLE OPERATING COSTS FOR A MID-SIZE SNOWMOBILE

	81/82	82/83	83/84	84/85	85/86
First Year					
Purchase	3,300	3,630	3,993	4,392	4,831
Spares	250	275	302	337	365
Fuel/6,000 km	696	894	1,044	1,212	1,392
TOTAL	4,246	4,769	5,339	5,941	6,588
¢/km	0.71	0.79	0.89	0.99	1.10
Fuel/12,000 km	1,392	1,728	2,088	2,424	2,784
TOTAL	4,942	5,633	6,383	7,153	7,980
¢/km	0.41	0.47	0.53	0.60	0.66
Second Year					
Overhaul	800	880	976	1,047	1,181
Spares	250	275	302	337	365
Fuel/6,000 km	696	864	1,044	1,212	1,392
TOTAL	1,746	2,019	2,322	2,596	2,938
¢/km	0.29	0.34	0.39	0.43	0.49
Fuel/12,000 km	1,392	1,728	2,088	2,424	2,784
TOTAL	2,442	2,883	3,366	3,808	4,330
¢/km	0.20	0.24	0.28	0.32	0.36
2 yr total/ 12,000 km	5,992	6,788	7,661	8,537	9,526
¢/km	0.50	0.57	0.64	0.71	0.79
2 yr total/ 24,000 km	7,380	8,516	9,749	10,961	12,310
¢/km	0.31	0.35	0.41	0.46	0.51

Table 7: SAMPLE OPERATING COSTS FOR A LARGE SNOWMOBILE

	81/82	82/83	83/84	84/85	85/86
First Year					
Purchase	5,000	5,500	6,050	6,655	7,260
Spares	300	330	363	399	438
Fuel/6,000 km	1,160	1,440	1,740	2,020	2,320
TOTAL	6,460	7,270	8,153	9,074	10,018
$/km	1.08	1.21	1.36	1.51	1.67
Fuel/12,000 km	2,320	2,880	3,480	4,040	4,640
TOTAL	7,620	8,710	9,893	11,094	12,338
$/km	0.63	0.73	0.82	0.92	1.03
Second Year					
Overhaul	900	990	1,089	1,198	1,318
Spares	300	330	363	399	438
Fuel/6,000 km	1,160	1,440	1,740	2,020	2,320
TOTAL	2,260	2,760	3,192	3,617	4,076
$/km	0.39	0.46	0.53	0.60	0.68
Fuel/12,000 km	2,320	2,880	3,480	4,040	4,640
TOTAL	3,520	4,200	4,932	5,637	6,396
$/km	0.29	0.35	0.41	0.47	0.53
2 yr total/ 12,000 km	8,820	10,030	11,345	12,691	14,094
$/km	0.73	0.84	0.94	1.06	1.17
2 yr total/ 24,000 km	11,140	12,910	14,824	16,731	18,734
$/km	0.46	0.54	0.62	0.70	0.78

Estimates are developed for:

- total running costs for a single year of operation for, 6,000 and 12,000 km.
- running costs for an overhauled machine in the second year of its life, for 6,000 and 12,000 km.
- total costs for buying a machine, retaining it, overhauled, for a second year of operation.

The estimates are tabulated in Tables 5, 6, and 7. The cheapest case is running a $1700 machine for two years over a total of 24,000 km (16,000 mi.); total costs then amortize at 0.20¢/km (0.32¢/mi.). These costs are projected to increase to 0.34¢/km (54¢/mi.) by 1986.

However, these machines are not in general use north of the tree-line. A more common case would be a mid-size machine, scrapped after a single year of use. With 6,000 km (3,750 mi.) covered during that year, the cost would total $4,246 or 71¢/km ($1.14/mi.); with 12,000 km (7,500 mi.), the total cost would be $4,942 or 0.41¢/km (0.64¢/mi.). This latter figure is interesting in that many respondents would say simply that the current working figure for annual operating costs has now reached $5,000. The estimates made here show that this figure will rise to about $8,000 by 1985/86.

Comparable figures for operating the large snowmobiles over the longer distances are already approaching $8,000 for a single year of use and will exceed $12,000 by 1985/86.

4. SNOWMOBILES AS WORKING VEHICLES IN THE ARCTIC: COMMENTS UPON PERFORMANCE AND SCOPE FOR IMPROVEMENT

In this chapter, the comments that mechanics, dealers and users made upon the performance of snowmobiles, and their suggestions for improvements, will be presented in relation to the four major design elements of the snowmobile: power plant, transmission, suspension, body.

The comments recorded during the field work tended to be directed either at snowmobiles, or certain snowmobile features, in general, or at particular makes or models. Since the circumstances did not permit us to give equal weight to all, or a representative selection of, machines during controlled operating conditions (as in consumer product testing), this account will avoid categorical comparisons between the various makes - unless the comments have some useful general application.

Each group of design features will be treated in the following way. The function will be briefly described, evolution since the 1960's will be summarised, user comments will be presented, and, finally, the scope for improvements will be estimated, taking into account the constraints of production economics.

4.2. POWER PLANT

The following elements of the power plant will be discussed in turn:

Engine
Cooling systems
Fuel supply & carburetors
Electrical system - ignition
Starter, manual, electric
Engine mounting

4.2.1. Engines

With one three-cylinder exception, all snowmobiles are powered by single or twin cylinder two stroke engines, with displacements ranging from 246 cc to 635 cc. With the exception of Bombardier, all engines are of Japanese manufacture. Bombardier engines are built by an Austrian subsidiary, Rotax.

Engine development has been governed both by the desire to improve power to weight ratios and the need to conform to the noise legislation introduced during the early 1970's. In this development, four strok and rotary engines were introduced on some models but never became popular.

Noise control influenced the trend away from the louder single cylinder engines towards smaller twin cylinder motors. This trend entailed the sacrifice of some torque, thus reducing the pulling power of snowmobiles.

Mechanics and dealers interviewed in this study were virtually unanimous in their view that the major cause of engine failure was improper care by owners. Not warming engines sufficiently was cited as the most common fault. Over-heating as a consequence of pulling over-laden komatiks was also felt to be a frequent cause of breakdown. Frequently, it was said, owners would not run in their new engines properly.

When asked what power of engine was most suited for general working use, most users preferred the 340 engines (engines with displacements in the range 338 to 368 are usually prefered to "340's"). A smaller group of users preferred 440 engines (these encompass the range 432 cc to 462 cc in current production models). Generally, those who preferred the smaller engine sizes felt that the power balance with the larger engines was unsuitable for pulling, especially from a standstill, when the tracks would spin and the machine would dig into the snow. But it should be noted that, often, those who stated a preference for the larger engines made specific reference to models which happened to have relatively large track areas and would thus be less imbalanced in the sense mentioned above.

When questioned about future engine development, manufacturers were understandably unwilling to reveal their plans, but it seems safe to state that a primary factor will be increasing fuel efficiency.

The multi-purpose models' recently introduced by Yamaha, Bombardier and John Deere all have 340 two cylinder engines. Next year, Bombardier will be offering a single cylinder 270 cc engine as an alternative in its Scandic model. The intention is to improve lugging power while reducing weight somewhat, and also reducing the price. Polaris will introduce a model aimed at the utility market; using a 440 engine de-tuned for fuel efficiency and geared down for pulling power.

4.2.2. Cooling systems

Three variations are in use: free air, fan-cooled, liquid cooled. At present, only one model, the John Deere Spitfire, is cooled by free air; the majority of the remainder are fan-cooled.

Free-air systems rely on the movement of the machine to create an air flow which is ducted over the engine. Fan-cooling simply employs a fan to produce the air flow, introducing a slight weight penalty and consumption of engine output. Liquid cooling permits engines to be pushed very hard but the weight, and cost, penalty is significantly higher than that for fan-cooling.

Early engines were usually cooled by free-air, but a demand for higher power and quieter engine noise led towards the introduction of liquid cooling. At this stage fan-cooled engines did not provide adequate noise reduction, partly because of poor design of cowlings and fans. Subsequent improvements in noise control technology have made fan-cooled engines more acceptable in this respect.

Users felt that fan-cooling provided a distinct advantage over the old free air systems, particularly when pulling loads at low speed, up hill, or when travelling through deep snow.

Though liquid cooling provided potential relief from such problems, most users felt that such advantages did not outweigh the added cost, weight and complexity of liquid cooling - and the extra horsepower produced was not required for pulling.

It can be concluded that current developments in the design of cooling systems work to the benefit of utility users. With increasing cost-consciousness amongst recreational users, improvements in the sound emission and efficiency of fan-cooling, it appears likely that, in future, liquid cooling will become increasingly reserved for only the very high performance machines.

4.2.3. Fuel supply and carburetors

Fuel tanks vary between 13.6 litres (3.0 gallons) and 30.7 litres (6.75 gallons). They are made out of plastic and moulded to conform to the shape of the vehicle. Most snowmobiles use a single carburetor, with the high performance types using two. All carburetors on current models are of Japanese manufacture.

Two systems are employed for mixing oil and gasoline. This is either pre-mixed by hand or mixed automatically by an oil injection system, which, besides accomplishing the mixture, is designed to improve performance by automatically adjusting the mixture ratio according to the performance of the engine.

There is evidence to show that the machines with oil injection use less oil than would be used when mixing oil and gas mixture (typically 50:1). However, one of the problems with current injection systems is that they are not easily recalibrated. Hence, for someone travelling a lot with their machine in different atmospheric conditions (high altitude) this can pose problems.

Fuel tanks and supply lines have become more durable and safer in the course of development. Carburetion has improved significantly, both in terms of fuel efficiency, smoothness of engine performance and ease of cold starting. Starting in very cold temperatures has been improved by the addition of primers. In the interests of reducing noise from carburetor air intake, air intake silencers have been developed which not only reduced noise but also improved performance and fuel economy.

The most significant design change has been the trend towards oil injection (75% of models in current production have this facility). For recreational use, or perhaps any form of use, this system has obvious practical advantages; drivers are relieved of the chore of mixing the fuels, carrying the separate containers necessary and trying to remember whether the oil is already in the gasoline or not. There is no question that this feature is a strong selling point for recreational vehicles.

But in regard to use in the Arctic, respondents tended to be equivocal. The major problem (apart from the approximately $75 cost penalty) is that at very low temperatures the oil would become too thick and could not be induced to mix with the fuel, with the result that the machines would be running on straight gasoline and the engines would seize.

This would start to happen as the temperatures fell below minus 20° C. Users would resolve it simply by reverting to pre-mixing and pouring the mixture directly into the fuel tank. But when there was still oil remaining in the oil injection reservoir, this would enter the fuel supply as the temperature rose, causing plugs to oil up. Some users simply remove the whole system.

The manufacturers responded to this problem by improving the design of the oil injection systems and by introducing oils that would be more miscible at low temperatures, (these are of course more expensive than regular oils). But still, many users do not feel inclined to trust oil injection systems, though, on the other hand, many do not feel there is any longer a real problem.

Discussions of the relative merits of oil injection and pre-mixing did not lead to any hard conclusions. It did seem, however, that, while for a recreational user an oil injection system may be a positive sales asset, Arctic utility users would not generally be turned away from a purchase simply because the machine did not possess oil injection.

Though oil injection is designed to improve performance as well as save labour, there was no indication from respondents that this was consistently perceptible enough to confer significant advantage over pre-mixing. All that can be said in conclusion is that, where lower costs, simple construction and operational security are at a premium, pre-mixed is superior; for labour-saving and perhaps a slight improvement in performance, oil injection is the better system.

The overall result of advances in fuel supply since the early days of snowmobile design has been to improve efficiency in consumption and operation and to improve the durability and safety of materials.

4.2.4. Electrical system: ignition

All current models employ capacitor discharge electronic ignition (CDI) systems, with the exception of the Bombardier Alpine and Elan, which retain contact breaker/magneto systems.

When the latter were common to all snowmobiles, ignition problems were among the more frequent frustrations visiting users, though these would be aggravated by problems with spark plugs, fuel mixture and the quality of wiring.

In current models, these have to a large extent been remedied, though when CDI was first introduced, the quality was generally poorer and breakdowns were more frequent than they are now. Users then experienced added frustrations due to the fact that such systems are less amenable to repair and adjustment on the trail than are breaker systems.

Mechanics and dealers on the whole felt that the reliability of CDI systems had improved to the point where their superior operational performance made them generally preferable to breaker systems.

4.2.5. Starters, manual, electric

About half the models currently in production rely upon manual starting only, most of the remainder offer both manual and electric while a few rely upon electric starting only.

Electric starting was widely regarded as inessential for utility use. It not only adds cost and complexity to the machines, but also suffers from battery problems in the extreme cold.

Users frequently complained about the quality of manual starter systems, particularly those containing synthetic materials. There is no doubt that, in the very cold periods, the bearing surfaces of manual starting mechanisms are subjected to more stress than in any other operating conditions.

In conclusion, though electrical starting systems may have improved during the late 1970's such systems as a whole are not required for utility usage. It should be pointed out that this conclusion applies to 340-440 utility machines; the Bombardier Alpine (635 cc) is in fact equipped with an electric only starting system. It appears that basically this is a marketing decision since most clients feel that on such a heavy machine the weight difference with electric start is not significant. However, there is no technical reason why it could not be offered manual only, apart from assembly line implications. The 640 engine is not that easy to crank; however, several years ago it was used in other model lines without electric start.

4.2.6. Engine mounting

With the exception of the Bombardier Elan and Alpine models, all snowmobile engines are now front-mounted, that is, they are placed over the skis rather than on the tunnel, which is the basic structural element of the snowmobile.

Only early snowmobiles, the engines were either tunnel mounted, just in front of the driver, or sometimes rear-mounted. Several factors influenced the gradual move forward and lower. Steering, especially on hardpack and prepared trails, became more positive due to the increased weight over the skis. Front-mounted engines can be more readily enclosed, making noise control more simple. Also, from the point of view of styling, a lower, front-mounted engine can be incorporated in a sleeker package than one mounted on the tunnel.

For travel on hardpack tundra snow surfaces, most respondents felt that the more positive steering provided by front mounted engines was a definite advantage, though an engine does not necessarily need to be low to the ground as well as far forward in order to achieve this effect. Users were less enthusiastic about low-mounted engines because of the increased likelihood of damage from impact while traversing uneven surfaces. Front-mounted engines are in addition easier to get at and work upon than those mounted on tunnels.

However, machines with front-mounted engines perform less agreeably in deep snow areas. One trapper, near Yellowknife described how he would lengthen the front struts so as to raise the hood above the skis. In such conditions, tunnel mounted engines confer a definite advantage.

Front mounting of the engine leads to an increase in cost of snowmobile construction since the attachment of the frame to the bottom plate must be redesigned to accept the additional stresses imposed by the engine mounting. The desire to produce a very sleek appearance further complicates the layout by requiring that the engine be mounted as low as possible. This results in a complex cross-member structure which joins the tunnel frame to the bottom plate and supports the engine mounting brackets. When aluminium is utilized instead of steel, the design and assembly become very sophisticated and it is rare for a new design not to encounter problems which require reinforcement the following year.

The advantages and disadvantages of this basic design change appear to come close to cancelling each other out. Certainly, for manufacturers to reverse this trend for utility snowmobiles, the advantages would have to be far more obvious than they are as such a change would entail considerable costs in re-tooling.

The disadvantages become more evident when deep snow operation is considered. There may be some scope here for developing a means of raising and lowering the front end as snow conditions vary. However, if the ski-legs were lengthened to provide a sort of "hydro foil" effect the problem would be the stress of the anchorages in the bottom plate since the effect of any impact on the skis would be significantly amplified by the longer moment arm.

4.2.7. Power plants: concluding remarks

Developments in engine efficiency have in general worked to the benefit of utility users, especially with respect to engine cooling, carburetion and ignition. Liquid-cooling and electric starting are redundant for utility use; oil injection, in its present stage of development has not yet proved to be consistently more reliable or efficient than pre-mixing. The forward displacement of power plants has proved to be acceptable for use on tundra snow; less so for use in deep, forest, snow.

4.3. TRANSMISSIONS

With the exception of the Bombardier Alpine (which has two forward gears and reverse) all snowmobiles utilise a variable drive system for transmissions of power to the track. The principle elements in this system will be discussed as follows:

Drive pulley (clutch), driven pulley, driven belt
Chain and chaincase
Drive axle and sprockets

4.3.1. Drive pulley (clutch), driven pulley, drive belt

Most snowmobiles utilize a form of automatic transmission or torque converter with a drive pulley or clutch linked to a driven pulley by a drive belt. This system is simple in concept, relatively light for the power transmitted, quite efficient and most flexible.